MODELING AND ANALYSIS OF THE PERFORMANCE OF A PARABOLIC TROUGH SOLAR CONCENTRATOR

By

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This Thesis was Submitted in Partial Fulfillment of the Requirements for the Master's Degree of Science in Engineering

Faculty of Graduate Studies

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May, 2011

The University of Jordan

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AKNOWLEDGEMENT

First of all, my full gratitude and thanks are to Allah the one who helped me and gave me life, power and time to finish this work.

My deep appreciation goes to many people whose advice, assistance and encouragement have enabled me to get to this stage in my life. I am really fortunate to have known so many great people so far in my life. To those who are missing in this brief list, and were supposed to be here, I sincerely apologize.

I would like to add a few warm words for the people who gave me support, directly or indirectly to complete this project. It was my pleasure that Prof. Mahmoud A. Hammad allowed me to work on this project with him. I must thank him for providing me all the necessary resources and help to finish this project. His experience in this field helped me a lot in finishing my work. I must say without his continuous help, guidance and support, this project would not have been successfully completed.

I also thank my parents and my family for their love and support.

Finally, but importantly, my wholehearted thanks go to my husband for his assistance and encouragement, I thank you from the bottom of my heart, and I will be forever grateful.

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LIST OF ABBREVIATIONS OR SYMBOLS

A_a : Collector aperture area (m^2).

CR: Concentration ratio.

C_{pw} : Specific heat of water (J/g.K).

D_{out} : Outer diameter of steel absorber (m).

h_{in}: The enthalpy at the inlet fluid temperature of the receiver (kJ/kg).

h_{out}: The enthalpy at the outlet fluid temperature of the receiver (kJ/kg).

I_s : Direct solar irradiance (kW/m^2).

l_a : Parabola width (m).

l : The absorber length(m).

$\dot{m}_w$: Water mass flow rate through the absorber tube of the collector (Kg/s).

P_{out} : Output power of the plant (kW).

$Q_{collector-ambient}$: The total thermal loss in a PTC (kW).

$Q_{absorber-ambient}$: Radiative heat loss from the absorber pipe to ambient (kW).

$Q_{absorber-glass}$: convective and conductive heat losses from the absorber pipe to its outer glass tube (kW).

T_{win}: The inlet temperatures of water (^oK).

T_{wout}: The outlet temperatures of water (^oK).

T_{amb}: The ambient air temperature (^oK).

$T_{absmean}$: The mean absorber pipe temperature (^oK).

ΔT_w : The change of water temperature (^oK).

T_c: The glass cover temperature (^oK).

U_l: The overall heat transfer coefficient (W/m^2K).

GREEK SYMBOLS

α : Absorptivity of the absorber selective coating.

ρ : Reflectivity of the collector reflecting surface.

γ : Intercept factor.

τ : Transmissivity of the glass tube.

η_g: Global efficiency.

η_{th} : Thermal efficiency.

η_s: Solar efficiency.

η_{opt}: optical efficiency.

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ABSTRACT

The increasing fossil fuel costs have led the world to use the free and naturally available energy from the Sun to produce electric power.

Parabolic trough technology is nowadays the most extended solar system for electricity production or steam generation for industrial processes. It is the most proven and lowest cost large-scale solar power technology available today. It is basically composed of a collector field which converts solar irradiation into thermal energy and a conventional thermal to electric conversion Rankine cycle. In these plants, a storage system can be implemented in order to increase plant production.

This work aims at finding a simulation model of a solar thermal power plant and suggests the use of the EuroTrough Concentrator as the solar collector of choice. MATLAB software is used for the analysis and performance evaluation of two different working fluids used in the simulation which are: water (as Pressurized and Boiling water), Oil (Therminol-VP1).

It was found that using water (pressurized or boiling) in the receiver tube is better than the Therminol-VP1 oil. And the pressurized water has the highest value of efficiency compared to the boiling water and Therminol-VP1 oil which has the lowest performance. Also the ET performance is applied in different places in Jordan, and the distribution of direct solar irradiance at different days around one year is calculate and plotted for Ma'an city.

Chapter One: Introduction

Solar energy is quite simply the energy produced directly by the sun and collected elsewhere, normally the Earth. Due to the nature of solar energy, two components are required to have a functional solar energy generator. These two components are a collector and a storage unit. The collector simply collects the radiation that falls on it and converts a fraction of it to other forms of energy (either electricity and heat or heat alone). The storage unit is required because of the non-constant nature of solar energy; at certain times only a very small amount of radiation will be received. At night or during heavy cloud cover, for example, the amount of energy produced by the collector will be quite small. The storage unit can hold the excess energy produced during the periods of maximum productivity, and release it when the productivity drops. In practice, a backup power supply is usually added, too, for the situations when the amount of energy required is greater than both what is being produced and what is stored in the container.

Methods of collecting and storing solar energy vary depending on the uses planned for the solar generator. In general, there are three types of solar collectors and many forms of storage units.

1.1 Types of solar collectors

The three types of solar collectors are:

1- Flat Plate Collector Types.

2- Evacuated Tube Collector Types.

3- Concentrated Collector Types.

1.1.1 Flat Plate Collectors

A typical flat-plate collector is a metal box with a glass or plastic cover (glazing) on top and a dark-colored absorber plate on the bottom as shown on Figure 1.1. The sides and bottom of the collector are usually insulated to minimize heat loss.

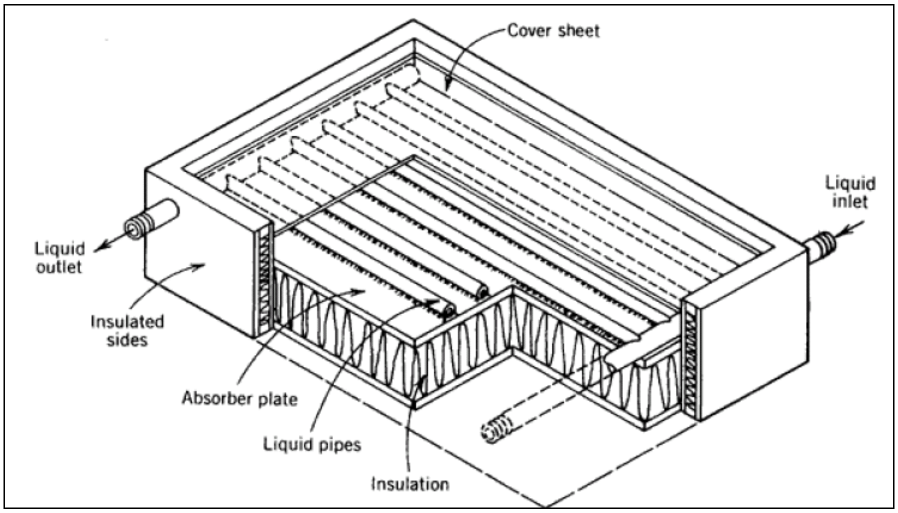


Figure 1.1 A typical flat-plate collector [Duffie J. and Beckman W. (2006)]

Sunlight passes through the glazing and strikes the absorber plate, which heats up, thus transforming solar energy into heat energy. The heat is transferred to liquid passing through pipes attached to the absorber plate. Absorber plates are commonly painted with "selective coatings," which absorb and retain heat better than ordinary black paint. Absorber plates are usually made of metal—typically copper or aluminum—because the metal is a good heat conductor. Copper is more expensive, but is a better conductor and less prone to corrosion than aluminum. In locations with average available solar energy, flat plate collectors are sized approximately one-half- to one-square foot per gallon of one-day's hot water use.

1.1.2 Evacuated-tube Collectors

Evacuated solar collectors (tubes) operate differently than the other collectors available on the market. Figure 1.2 describes the evacuated tube collector. The basic idea is that the solar collectors consist of a heat exchanger inside a vacuum-sealed tube. The collectors are usually made of parallel rows of transparent glass tubes. Each tube contains a glass outer tube and metal absorber tube attached to a fin. The fin is covered with a coating that inhibits radiative heat loss. Air is evacuated from the space between the two glass tubes to form a vacuum, which eliminates conductive and convective heat loss. The vacuum tube envelope therefore minimizes heat loss and ensures high collector durability and steady performance. Figure 1.3 shows one version the vacuum tube collectors.

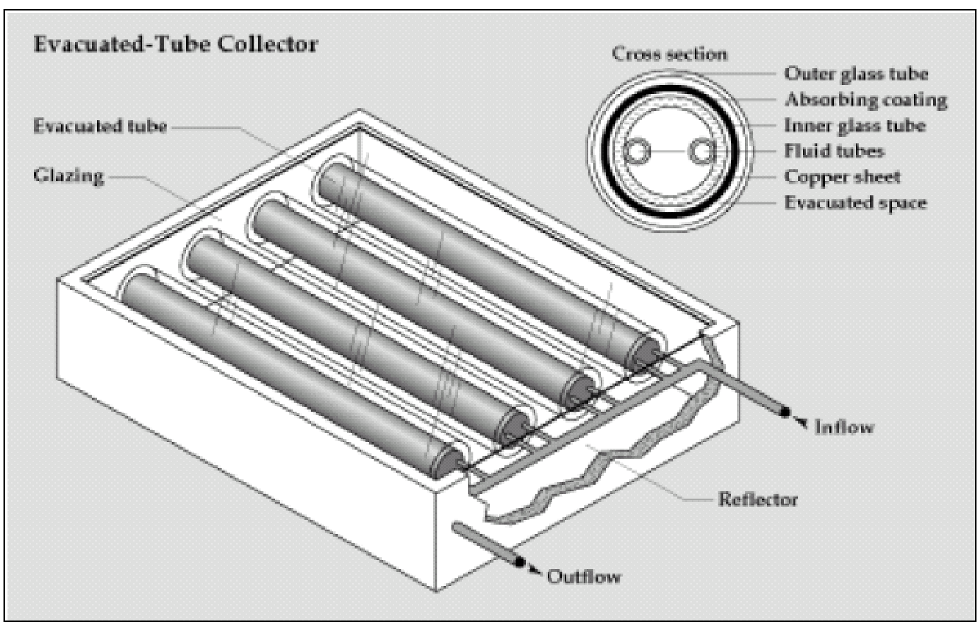


Figure 1.2: Evacuated-tube collectors are efficient at high temperatures [Duffie J. and Beckman W. (2006)].

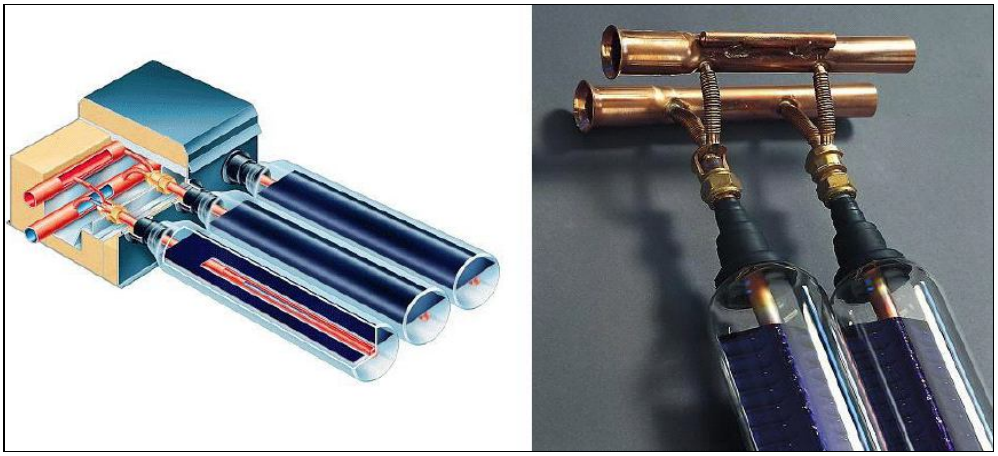


Figure 1.3: High efficiency vacuum tube collectors [Kreith K and Goswami Y. (2007)]

The design features a vacuum contained between two concentric glass tubes, with the absorber selective coating on the inside tube. Water is typically allowed to thermosyphon down and back out the inner cavity to transfer the heat to the storage tank. There are no glass-to-metal seals. As one could imagine, evacuated tube collectors are more expensive that flat-plate collectors, on an approximate 2:1 ratio per unit of surface although this type of evacuated tube has the potential to become cost-competitive with flat plates. Applications: The temperatures that can achieved are much higher (~80-180ºC), making them more appropriate for industrial or cooling applications.

1.1.3 Concentrated Collector Types:

In these models, as opposed to the ones included before, the sun rays are concentrated into one designated area. This entails higher temperatures and broader uses for the heat achieved as the energy also increases.

A- Parabolic Trough

A parabolic trough is constituted by a long parabolic mirror through which focal point runs a Dewar tube. Due to their parabolic shape, this mirror reflects and concentrates the

sunlight directly upon the receiver tube, focusing the sun at ~80 times its normal intensity. The trough is usually aligned on a north-south axis, and rotated to track the sun as it moves across the sky each day as shown in Figure 1.4. Alternatively the trough can be aligned on an east-west axis. This reduces the overall efficiency of the collector, due to cosine loss, but only requires the trough to be aligned with the change in seasons, avoiding the need for tracking motors. In any case, the latter are used to increase efficiencies. The concentrated energy is absorbed by a heat transfer fluid (HTF), usually oil that flows through the Dewar tube. This fluid is then used to generate steam which powers a standard turbine generator.

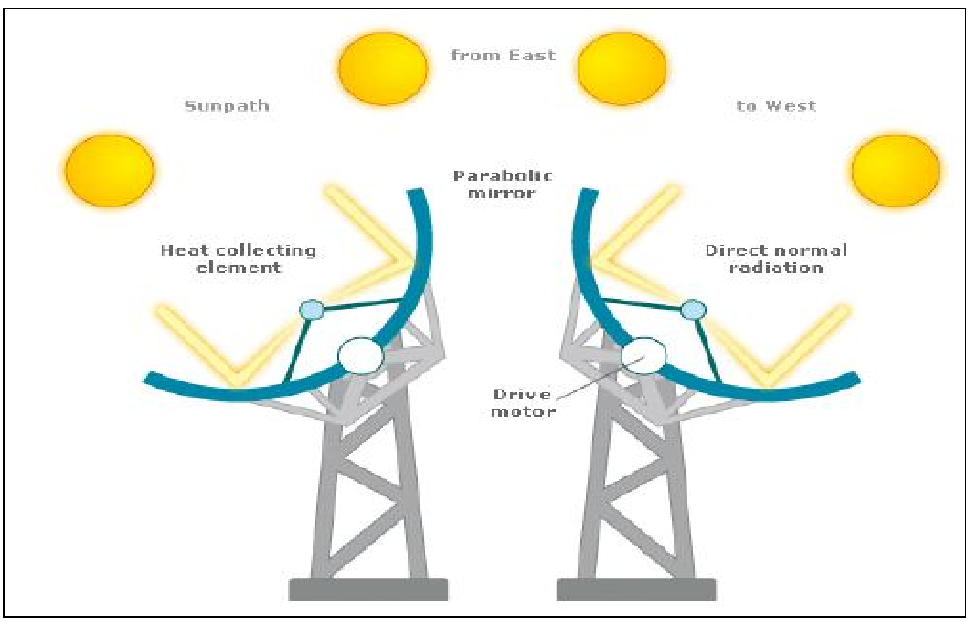


Figure 1.4: Parabolic trough following sun path [Mammadov F., Samadova U., and Salamov O. (2008)]

The process itself is economical and, for heating the pipe, thermal efficiency ranges from 60 to 80%. The overall efficiency from collector to grid, i.e. (Electrical Output Power)/(Total Impinging Solar Power) is about 20-25%. The basic component of a parabolic trough solar field is the solar collector assembly or SCA. A solar field consists of hundreds or potentially thousands of solar collector assemblies. Each solar collector

assembly is an independently tracking, parabolic trough solar collector composed of the following key subsystems:

- Concentrator metal support structure
- Mirrors or reflectors
- Linear receiver or heat collection element
- Tracking systems

The initial installation cost of parabolic trough solar collectors constitutes the main ingredient in the final cost of electricity produced by the power plant. And the main sources of cost (over 80%) of the collectors are the following: the metal support structure, the parabolic mirror, the receiver (heat collector element, HCE), and the tracking system

Concentrator metal support structure

This is the structure, usually composed of steel beams and rods, which gives and maintains the parabolic form of the mirrors as shown in Figure 1.5. Because the mirror itself is not geometrically rigid, the rigidity of the parabolic form wholly relies on the support structure. The technical difficulty is important due to the requirements on optical precision and wind resistance. Despite the heavy structure (up to 20kg per m2 of opening), material is only a very small fraction of the total cost. This structure needs careful design, and has to be assembled, installed and aligned on the field. All these explain the high labor cost.



Figure 1.5: Parabolic concentrator metal support structure [Cansee S., Pattiya A., Pattanasethanon S. and Sombuttera W., (2010)].

One key point of the design of the structure is that it has to have extreme tolerances and structural rigidity in order to focus the mirrors precisely on the receiver pipe. High resistance to large wind loadings on the large structure is a very important parameter.

Mirrors or reflectors

The parabolic mirror shown in Figure 1.6 is pieced up from several curved glass mirrors fixed on the support structure. Again, it is the production cost that largely dominates, with the obvious technical difficulty of forming the curved glass with precision. This inevitably limits the concentration ratio and the efficiency of the collector.



Figure 1.6 Parabolic mirrors [Pérez C., Porras-Soriano A. and Soriano M. (2009)]

Development efforts are aimed at reducing the thickness of the mirror, improving the reliability of the glass to metal seal, surface coatings on the mirrors to improve their performance and development of a composite concentrator modules with lightweight, front-surface mirrors instead of heavy (4 mm) glass mirrors that were originally used on the original SEGS plants. Industrial Solar Technology Company (IST) has developed a concentrator system that incorporates the reflective surface as part of the collector structure. This results in a very lightweight, low-cost concentrator module that is also very strong.

Linear receivers or heat collection elements (HCE)

The heat collection element (HCE) is composed of a stainless steel pipe with a glass tube surrounding it, with the space between evacuated to provide low thermal losses from the pipe, as shown in Figure 1.7. The pipe is coated with a material that improves the absorption of solar energy. The receiver collects the solar radiations reflected by the mirror, and transforms the energy into the heat conveyed out by the heat transfer fluid

(HTF) that flows in the inner tube. It is the most sensitive part of the trough, with two main difficulties: the selective coating with maximal absorptivity and minimal emissivity, and the metal to glass hermetic sealing to maintain the vacuum between the two tubes. The HCE is also a huge maintenance problem, with its high rate of failures, leaks and breaks. The cost of the selective coating, as well as radiative heat loss, should be independent to the size of the trough. On the other hand, the cost of the sealing will clearly increase if the size of the trough is decreased. Conductive heat loss will also increase with decreasing size (see the section on the receiver).

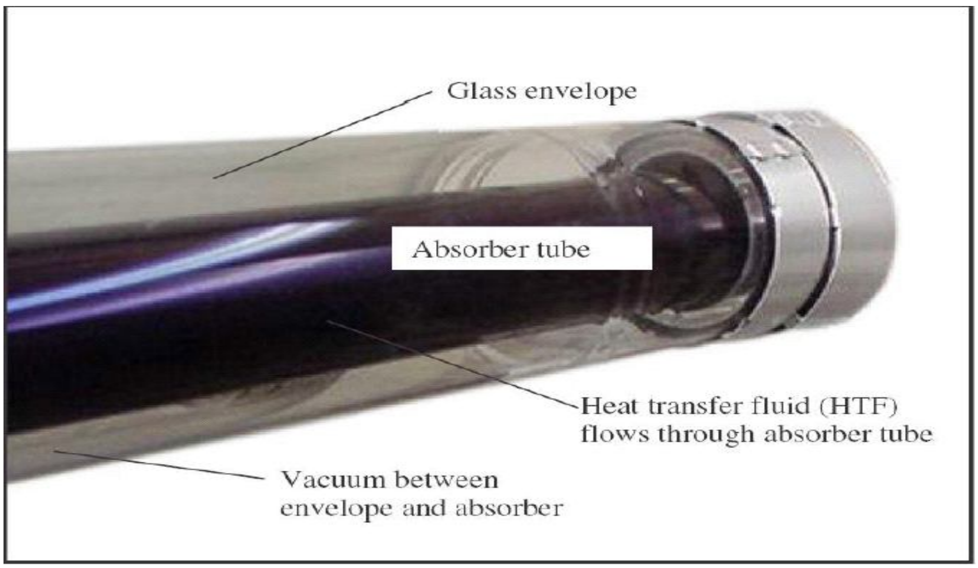


Figure 1.7 Heat Collection Element (HCE) [Mammadov F., Samadova U., and Salamov O. (2008)]

Tracking systems It is the mechanism that constantly maintains the orientation of the mirrors towards the moving sun. This part clearly has a huge cost reduction potential. The Plataforma Solar de Almería (PSA), the European Test Center for Solar Energy Applications, in southeastern Spain is Europe's test facility for concentrating solar power development. They are developing direct steam generation technology to eliminate the thermal oil system that is currently used in solar trough plants, which would greatly

simplify the plants. PSA is also developing new trough collector assemblies and coatings to improve the absorption of solar energy on the receiver tubes and to decrease the reflectivity of glass.

B- Solar Tower/Central Receiver

In this concentrating solar power (CSP) technology, there is a single receiver placed on top of a tower surrounded by hundreds of large, flat, mirrors (heliostats) which follow the apparent motion of the sun in the sky and which re-direct and focus the sunlight onto the receiver, as in Figure 1.8. The heliostats are provided with a two-axis tracking system. A heat-transfer fluid heated in the receiver is used to generate steam, which, in turn, is used in a conventional turbine generator to produce electricity, Figure 1.9 shows the schematics of a power tower plant. Some power towers use water/steam as the heat-transfer fluid. Other advanced designs are experimenting with molten nitrate salt because of its superior heat-transfer and energy-storage capabilities. If a gas or even air is pressurized in the receiver, it can be used alternatively to drive a gas turbine (instead of producing steam for a steam turbine). For gas turbine operation, the air to be heated must first pass through a pressurized solar receiver with a solar window.



Figure 1.8: Power tower power plant [Duffie J. and Beckman W. (2006)].

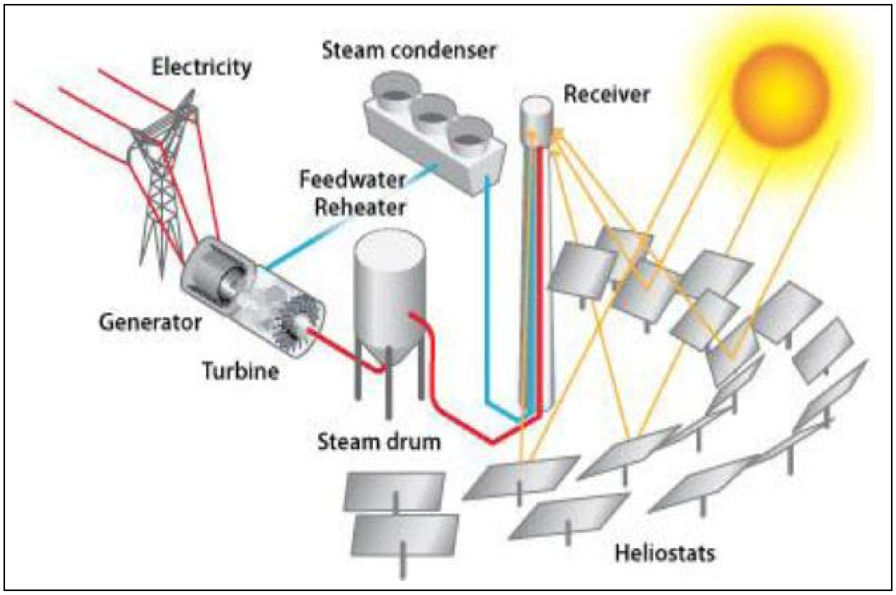


Figure 1.9: Schematics of a power tower plant [Duffie J. and Beckman W. (2006)]

C- Parabolic Dish/Stirling Engine:

A dish Stirling system consists of a parabolic-shaped point focus concentrator in the form of a dish that reflects solar radiation onto a receiver mounted at the focal point. These concentrators are mounted on a pedestal and can pivot on two axes to follow the sun. This two-axis tracking mechanism allows the capture of the highest amount of solar energy at any time possible, Figure 1.10 shows dish stirling power plant. The energy absorption enables fluid in the receiver to be heated to approximately 750°C. This is then used to generate electricity. Like all concentrating systems, they can be additionally powered by natural gas or biogas, providing firm capacity at any time.



Figure 1.10: Dish Stirling power plant [Mammadov F., Samadova U., and Salamov O. (2008)]

It is a concentrating solar power (CSP) technology that produces relatively small amounts of electricity compared to other CSP technologies. Overall size typically ranges from 5 to

15 meters in diameter and 5 to 50kW of power output, with a peak efficiencies of up to 30% (net) Figure 1.11 shows the schematics of a parabolic dish/stirling engine system.

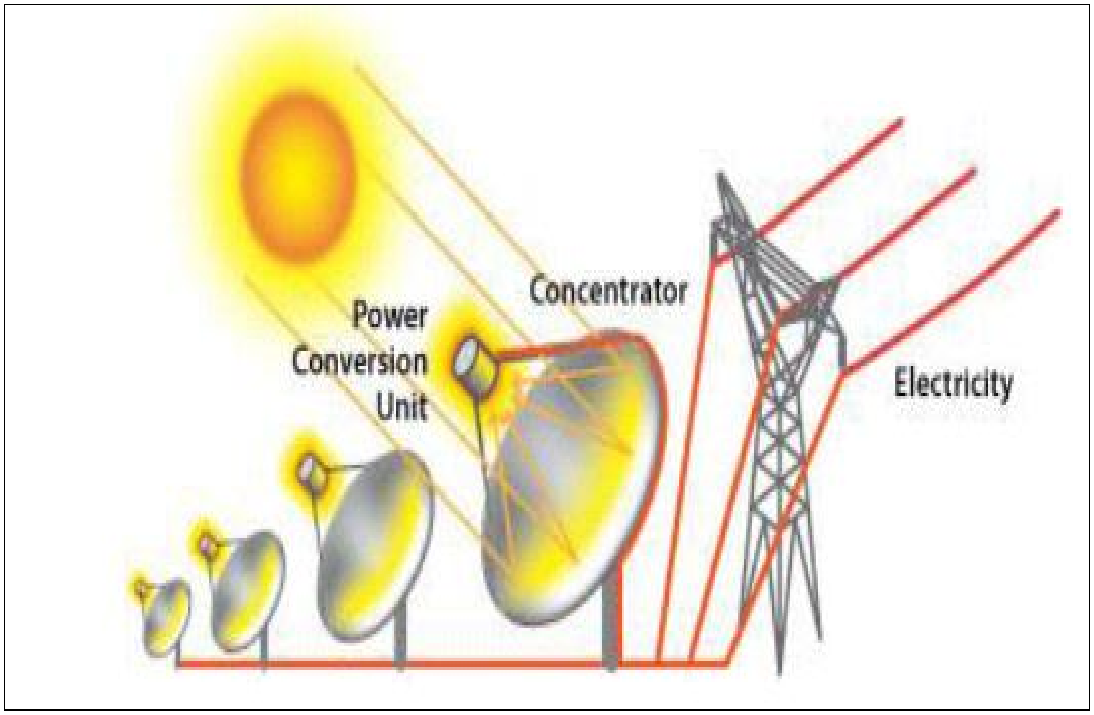


Figure 1.11: Schematics of a parabolic dish/Stirling engine system [Cimat A. (2001)].

D- Fresnel Mirror Concentrator

A Linear Fresnel Reflector (LFR) is a single-axis tracking technology that focuses sunlight reflected by long heliostats onto a linear receiver to convert solar energy to heat. The classical linear Fresnel system uses an array of mirror strips close to the ground to direct solar radiation to a single, linear, elevated, fixed receiver. Each mirror element is tilted at an angle so that all incident solar rays falling on them are reflected to a common focus. The Fresnel reflector is composed of parabola slices mounted on a flat surface. The flat mounting surface has advantages with regard to practical engineering and construction, Figure 1.12 shows the Fresnell geometry using three co-focal parabolas.

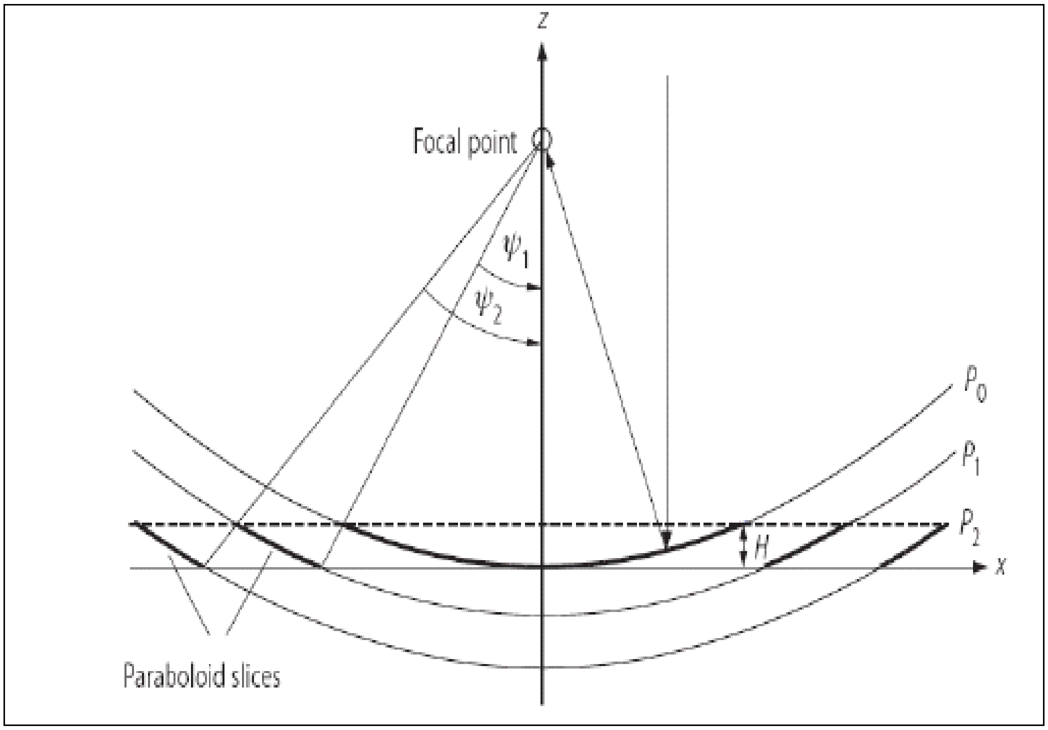


Figure 1.12: Fresnell geometry using three co-focal parabolas [Colina J., Lopez A. and Machuca F., (2010)]

Compact linear Fresnel reflector (CLFR) technology is a new configuration of the Fresnel Reflector field that overcomes the problem of reflector spacing, as shown in Figure 1.13. Traditional LFR technology design is based around one absorber tower.



Figure 1.13: Array of mirror strips of a CLFR [Cansee S., Pattiya A., Pattanasethanon S. and Sombuttera W., (2010)].

Chapter Two: Literature Review

As interest for clean renewable electric power technologies grows, a number of parabolic trough power plants of various configurations are being considered for deployment around the globe. It is essential that plant designs be optimized for each specific application. Different authors have provided techniques to enhance the performance of the parabolic trough collectors, some of them are mentioned below:

1) **Lippke (1995)** produced a detailed thermodynamic simulation model of the Solar Electric Generating Systems (SEGS VI) and power cycle using EASY simulation software. The objective of this model was to simulate system behavior during part-load conditions (such as winter months and cloud covered days). In this model, design state points from the technical plant description are used to back-calculate turbine state efficiencies and overall conductance values for all heat exchangers in the cycle. As results model was validated against hourly plant data for both a clear summer day and a clear winter day.

2) **Patnode (2006)** made a model for the solar field using the TRNSYS simulation program. The system collect energy using a synthetic heat transfer fluid pumped through absorber tubes in the focal line of parabolic trough collectors. The heated fluid provides the thermal resource to drive a Rankine steam power cycle. The Rankine power cycle was separately modeled with a simultaneous equation solving software. The steady-state power cycle performance was regressed in terms of the heat transfer fluid temperature, heat transfer fluid mass flow rate, and condensing pressure, and implemented in TRNSYS. As results, the gross system

efficiency was found to be insensitive to variations in the mass flow rate and solar field outlet temperatures within the observed operating range of the plant.

3) **Brooks (2005)** developed a parabolic trough collector, similar in size to smaller scale commercial modules, locally for use in an ongoing PTSC research programme. The aim of this study was to test and fully characterize the performance of the collector.

Specialized logging software was developed to record test data and monitor PTSC performance in real time. Two heat collecting element were tested with the collector, one unshielded and the other with an evacuated glass cover. Testing was carried out according to the ASHRAE 93-1986 (RA 91) standard, yielding results for the thermal efficiency, collector acceptance angle, incidence angle modifier and collector time constant. Peak thermal efficiency was 55.2% with the unshielded receiver and 53.8% with the glass shielded unit.

4) **Mammadov et al (2008)** investigated the utilization of parabolic-trough solar plant in the oil industry which is a relatively new application in the area of solar energy usage. The oil fields are utilized for the purpose of crude oil treatment. As results, a new method of a solar energy application has been highlighted. Also new technical and technological opportunities of a solar energy application in crude oil treatment in oil fields has been established and improved practical engineering on a solar thermal application.

5) **Brooks et al (2006)** introduced the performance of a South African parabolic trough solar collector (PTSC) module using the ASHRAE standard. The collector is designed for component testing and development in a solar energy research

programme. The primary aim of this study was to establish baseline performance of a PTSC research module, built at Mangosuthu Technikon and tested. Results will be used in an ongoing solar research programme to quantify improvements to concentrating collector components, tracking methodologies, operating techniques and the test facility's data acquisition system.

6) **Lüpfert et al (2001)** designed a new support structure of the collector included concept studies, wind tunnel measurements, finite elements method (FEM) analysis and resulted in a structure with a central framework element. This torque box design will have lower weight and less deformation of the collector structure than the other designs considered. Therefore it will be possible in future to connect more collector elements on one drive which results in reduced total number of drives and interconnecting pipes, thus reducing the installation cost and thermal losses. The presented design has a significant potential for cost reduction, the most important goal of the EUROTROUGH project. The prototype has been set-up and is under testing at PSA (Plataforma Solar de Almería) for its thermal and mechanical properties.

7) **Fischer et al (2006)** tested the collector efficiency of a parabolic trough collector prototype according to the European Standard EN 12975. The Standard includes, apart from the well-known steady state parameters, an incident angle modifier for diffuse irradiation and an effective collector thermal capacity. The addition of these two collector parameters allows the evaluation of continuous measurements over several hours even under irradiance fluctuations and changing sun position. As a result the efficiency of parabolic trough collector prototype has been tested

according to EN Standard 12975 using the test method under quasi dynamic conditions. This test method allows varying ambient conditions and continuous measurements over the day. The results show a very good agreement between measured and modeled collector output.

8) **Ciemat et al (2001)** Developed an advanced parabolic trough collector for various applications in the 200-400°C temperature range for solar fields capable of delivering up to hundreds of MW of energy. The project results more competitive solar power generation and achieved based on the expectations of lower cost and higher performance of the collector.

9) **Pérez et al (2009)** investigated the use of low and medium temperature active solar energy systems for the disinfestations of greenhouse soils. Four flat plate solar collectors (low-temperature solar thermal energy devices) and six parabolic trough concentrators (medium-temperature solar thermal energy devices) were used to heat water, which, via a buried heat exchange system, was used to heat the soil of greenhouse plots. Authors conclude that the soils heated by the flat plate solar collectors accumulated less energy than those heated by the parabolic trough concentrators and so the flat plate solar collector systems showed a better energy performance than the parabolic trough concentrator systems.

10) **Al-Soud et al (2008)** proposed a prototype of a 50 MW concentrating solar power plant (CSPP) for electricity generation in Jordan and performed the analysis of its economic feasibility. And a calculation model using the concept design of the proposed CSPP, and the solar irradiation data was developed to estimate the energy yield of the plant.

11) **Colina et al (2010)** developed a mathematical model which allows simulating the reflection of direct solar radiation on a CPC. Using analytical geometry and vector calculation, equations were evaluated, first for calculating the Cartesian coordinates of the reflecting surface. Later these points are used to calculate incident and reflected rays layouts on the collector. Visual Basic (Excel environment) program was developed for data generation and plotting the reflected rays at any instant. The incident radiation on the receptor was plotted separately displaying the daily direct energy distribution in the absorber. This mathematical model can be a first approach for absorption models which include direct solar radiation in photochemical or photothermal applications. The simulations with three reflectivity values showed that higher reflective materials give more uniform energy distribution, being more convenient for photochemical or photothermal applications which use solar radiation as energy source. Lower reflectivity values give poor incident energy distribution at rear zones of the absorber.

12) **Geyer et al (2002)** developed the high-performance EuroTrough parabolic trough collector models ET100 and ET150 for the utility scale generation of solar steam for process heat applications and solar power generation. A test with photogrammetric surface analysis has been performed. Its results served for detailed 3- dimensional insight in the reflector and support structure properties and improvements towards higher collector efficiency.

In this work the performance of a solar trough concentrator as a source of energy for electrical generation will be studied and analyzed. The parabolic-trough collector selected for the analysis is the EuroTrough collector. In this work, the heat transfer fluid in the receiver (oil) will be replaced firstly by pressurized water and then boiling water. In the end, the aspect of generation of electrical power from the solar plant is explained along with calculations and simulation. The simulation has been done in MATLAB through writing a program code. A comparison will be made between the obtained results of Euro-Trough and other existing parabolic trough collectors.

Chapter Three: Description of Power Plant

3.1 Introduction

At present, parabolic trough plants represent the most efficient and economical technology available to generate electricity from solar energy. This technology has been employed in large-scale installations for more than 20 years, showing its impressive potential on a day-to-day basis.

A parabolic trough plant consisting of a solar field and a conventional power block produces environmentally-friendly solar electricity within a scope of 30 to more than 250 megawatts. This will supply anywhere from 100,000 and 1,000,000 people with electrical energy in a safe, reliable and clean way.

Parabolic trough-shaped mirrors in the solar field concentrate the solar radiation onto an absorber tube located in the focus line of the collector. A heat transfer fluid inside the absorber tube is heated by the absorbed solar radiation and produces steam in the power block by means of heat exchangers. From there, the steam is fed into a turbine to generate electricity the way it is done in conventional power plants. With an integrated thermal energy storage system, the supply of electricity can be stored.

3.2 Parabolic Trough Power Plant Technology

Parabolic trough power plants consist of large fields of parabolic trough collectors, a heat-transfer fluid/steam generation system, a Rankine steam turbine/generator cycle, and thermal storage or fossil-fired backup systems (or both). These systems are illustrated schematically in Figure 3.1.

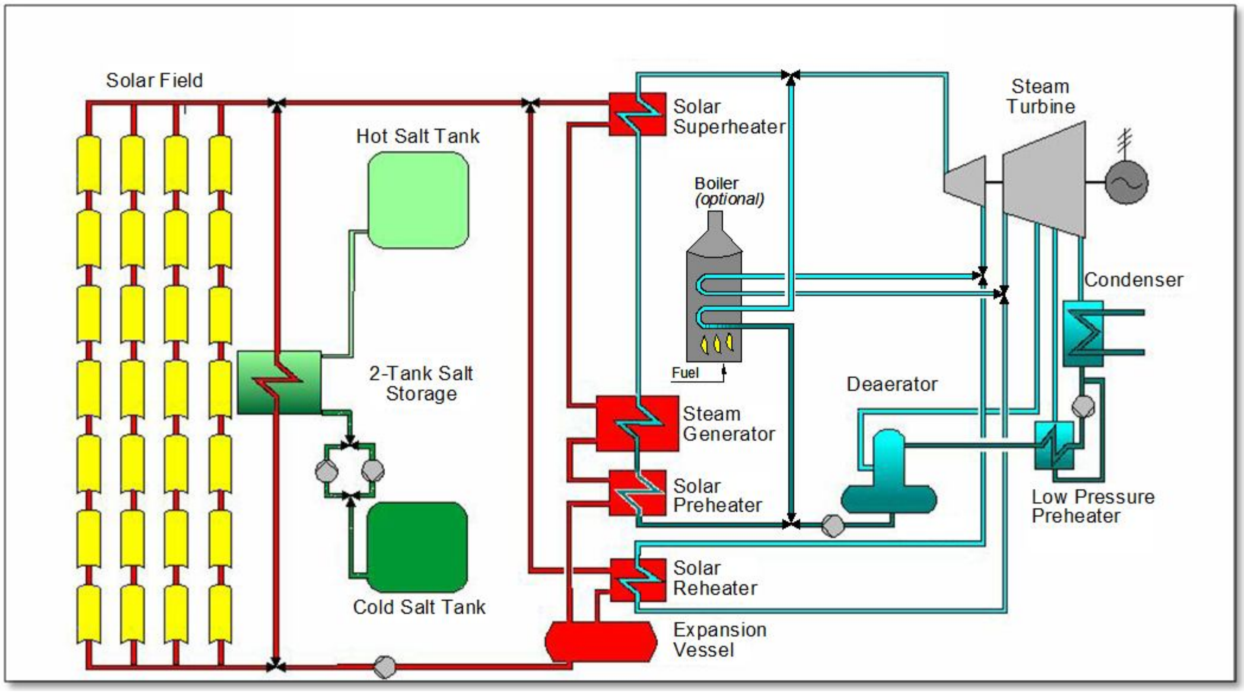


Figure 3.1: Schematic Flow Diagram of Parabolic Trough Plant [Pérez C., Porras-Soriano A. and Soriano M. (2009)]

The technology can be described as follows. The solar field is modular in nature, and it comprises many parallel rows of solar collectors aligned on a north-south horizontal axis. The linear parabolic-shaped reflector in each solar collector focuses the sun's direct beam radiation on the linear receiver at the focus of the parabola as seen in Figure 3.2. The collectors track the sun from east to west during the day to ensure that the sun is continuously focused on the linear receiver. A heat transfer fluid is heated to 391ºC as it circulates through the receiver and returns to a series of heat exchangers in the power block, where the fluid is used to generate high-pressure superheated steam (100 bar, 371ºC). The superheated steam is then fed to a conventional reheat steam turbine/generator to produce electricity. The spent steam from the turbine is condensed in a standard condenser and returned to the heat exchangers via condensate and feed-water pumps to be transformed back into steam. Condenser cooling is provided by

mechanical draft wet cooling towers. After passing through the HTF side of the solar heat exchangers, the cooled HTF is re-circulated through the solar field.

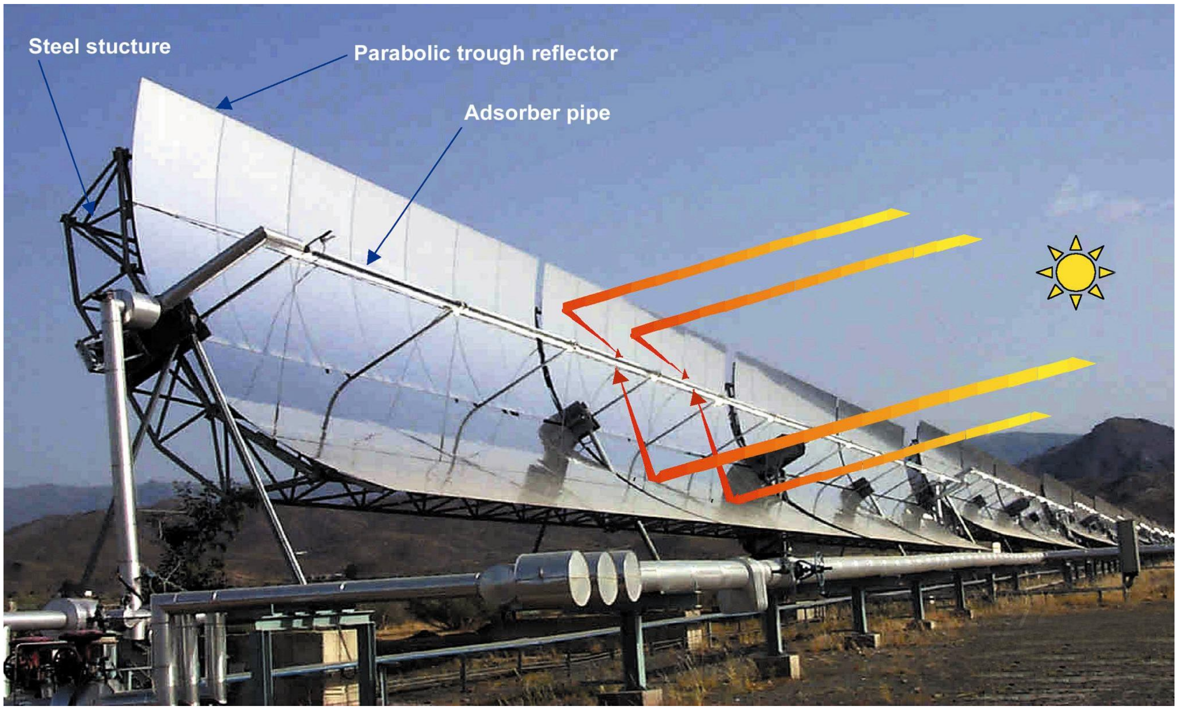


Figure 3.2: Parabolic Trough Collector [Paal R. Milow R. Ferriere A. Steinfeld A. and Zarza E. (2007)].

3.3 Trough Power Plant Performance Characteristics

Solar thermal electric power plants are designed to harvest available sunlight, either converting it to electricity immediately, or storing it for future use. The ability to store collected thermal energy is particularly important and can lead to a solar-only power plant with firm dispatching capability. Significant flexibility exists to design plants to provide specific energy services, that is, MWe capacity, ability to meet peak loads, and tailored annual energy production. Design tools have been developed that determine the proper collector field capacity, energy storage capacity, and turbine size to produce the required energy, given expected solar insulation and climatic conditions at each site. The evaluation is dominated by the variability of the solar resource. Some of the relevant data

are totally predicable (such as when the sun will rise on a given day), but most of the information depends on historically valid weather data.

3.4 EuroTrough collector

The parabolic-trough collector selected for the analysis is the EuroTrough collector, which was developed by a European consortium and is still under investigation at the Plataforma Solar de Almería (PSA) facilities. EuroTrough collectors can provide heat in a temperature range of 200- 400ºC for a variety of applications such as industrial process heat and electricity generation. The 98.5-m-long model was selected as the heat supply unit. Figure 3.3 shows the EuroTrough collector model chosen for the simulation and Table 3.1 lists its characteristics and dimensions.



Figure 3.3: View of the EuroTrough support structure [Brooks M. Mills I. and Harms T. (2006)].

Table 3.1: Parameters of the ET-100 Parabolic Trough Collector

Overall length of a single collector (m)	98.5
Number of parabolic trough modules per collector	8
Gross length of every concentrator module (m)	12.27
Parabola width (m)	5.76
Outer diameter of steel absorber pipe (m)	0.07
Inner diameter of steel absorber pipe (m)	0.055
Number of ball joints between adjacent collectors	4
Net collector aperture per collector (m2)	548.35
Peak optical efficiency	0.765
Cross section of the steel absorber pipes (m2)	$2.4*10^{-3}$
Inner roughness factor of the steel absorber pipes (m)	$4.0*10^{-5}$
Relative roughness of the steel absorber pipes (m)	$7.23*10^{-4}$

The EuroTrough collector models are made up of identical 12.27m long collector modules. Each module comprises 28 parabolic mirror panels. Each mirror is supported on the structure at four points on its backside. This permits the glass to bend within the range of its flexibility without effect on the focal point. The ET collector has 8 collector modules and an aperture area of 545m^2.

The following formula describes the trough surface equation:

$$Y = \frac{X^2}{4P}$$

(3.1)

Where: Y is the distance from the baseline, X is the distance from the Y axis and P is the focal distance, or the Y coordinate of the focal point or focal length.

3.5 Linear Receiver or Heat Collection Element

The radiating solar energy is directed to the receiver, absorber, or heat collection element (HCE) by the collector array, a typical receiver tube of a Parabolic Trough collector is shown in Figure 3.4.

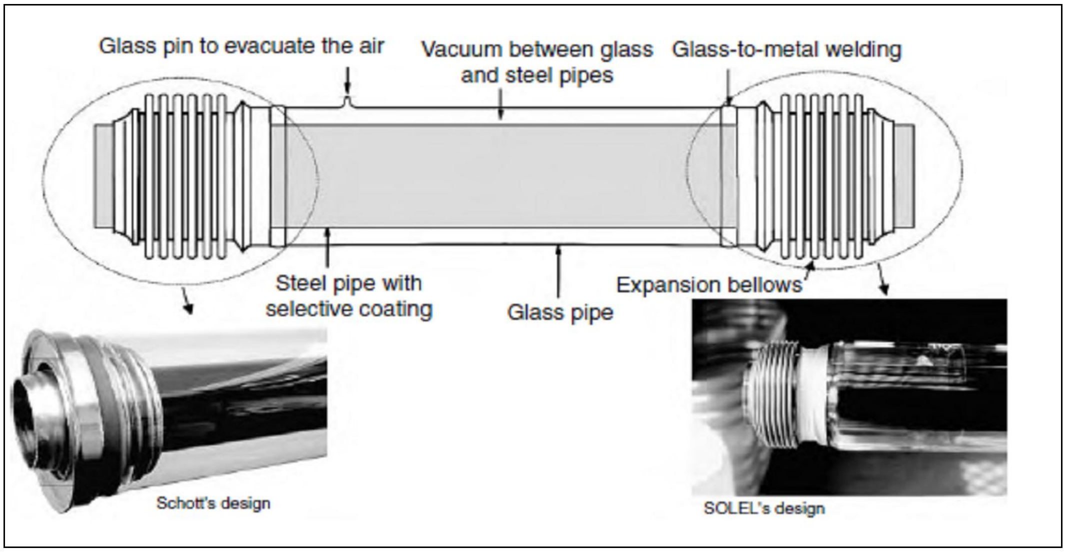


Figure 3.4: A typical receiver tube of a parabolic trough collector [Duffie J. and Beckman W. (2006)]

The receiver is a 4-meter-long, 70-mm diameter stainless steel tube with a special solar-selective absorber surface, surrounded by an anti-reflective evacuated 115-mm diameter glass tube. Located at the mirror focal line of the parabola, the receiver heats a special heat transfer fluid as it circulates through the receiver tube.

The receiver has glass-to-metal seals and metal bellows to accommodate for differing thermal expansions between the steel tubing and the glass envelop. They also help achieve the necessary vacuum-tight enclosure.

The vacuum-tight enclosure primarily serves to significantly reduce heat losses at high-operating temperatures. It also protects the solar-selective absorber surface from oxidation.

The selective coating on the steel tube has good solar absorptance and a low thermal emittance for reducing thermal radiation losses. The glass cylinder features an anti-reflective coating to maximize the solar transmittance. Getters—metallic compounds designed to absorb gas molecules—are installed in the vacuum space to absorb hydrogen and other gases that permeate into the vacuum annulus over time.

3.6 Heat Transfer Fluid (HTF)

The HTF can be one of the following: water, water-glycol mixtures, silicone compounds, fluorocarbons, inorganic salt mixtures, hydrocarbons, or liquid metals.

If other than water is used, it is conducted through a heat exchanger to produce superheated steam to propel a turbine or heat engine. Oils are often used as a HTF, but water and molten salts are gaining special attention in the solar thermal power plants realm.

3.6.1 Water

Water is nontoxic, nonflammable, inexpensive, and has excellent heat transfer characteristics. Direct Steam Generation (DSG) systems are currently under study.

Water will be used instead of oil in my study because of the following reasons:

- Higher steam temperatures can be attained compared to oil.
- Overall plant configuration is simpler than oil.
- Lower investment and O&M costs compared to oil.
- Higher plant efficiency than oil.

In this study water will be used as: pressurized water (water under high pressure to maintain it liquid under high temperatures) or as b) boiling water (water under normal pressure which transform to steam under high temperatures).

Chapter Four: Mathematical Model and Computer Simulation

4.1 Introduction

In this chapter, a mathematical model for parabolic trough collector will be described. The model is for estimation of energy outputs, losses and efficiencies of various parts of the system when the system is operating under different condition. A model simulating program of ET system has been developed by MATLAB software.

4.2 Performance parameters and losses of ET collector

One of the design parameters required for a ET collector is the geometric concentration ratio which is the ratio between the collector aperture area and the reflected area of the absorber tube it is given by:

$$CR = \frac{A_a}{A_r} = \frac{l_a l}{0.5\pi\, D_{out} l} = \frac{l_a}{0.5\pi D_{out}}$$

(4.1)

$$CR = \frac{5.76}{0.5 * \pi * 0.07} = 52.4$$

Where: CR is the Concentration ratio, A_a is the aperture area, A_r is the receiver area, l is the absorber length, l_a is the parabola width and D_{out} is the outer diameter of steel absorber. Figure 4.1 shows the design parameters required for a parabolic trough collector.

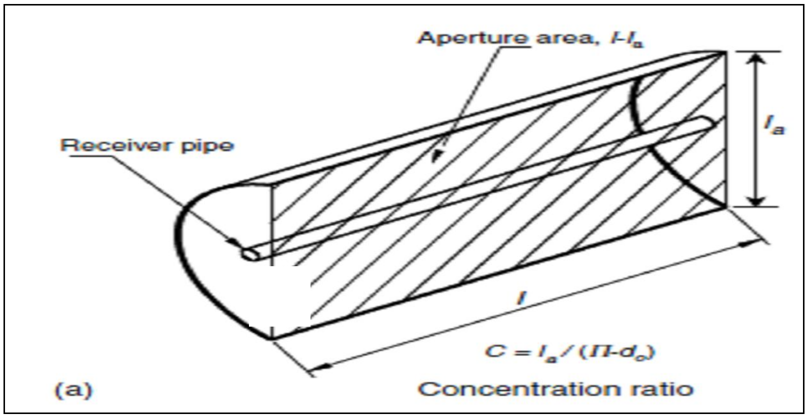


Figure 4.1: Concentration ratio of PTC [Duffie J. and Beckman W. (2006)].

When direct solar radiation reaches the surface of a PTC, a significant amount of it is lost due to several different factors. The total loss can be divided into three types, which in descending order of importance are:

- Optical losses.
- Thermal losses from the absorber pipe to the ambient.
- Geometrical losses.

The optical losses can be expressed by the optical efficiency which is given by:

$$\eta_{opt} = \rho * \gamma * \tau * \alpha$$

(4.2)

=0.78 for typical values of ρ, γ, τ *and* α. $= 0.93 * 0.95 * 0.93 * 0.95$

Where: ρ is the reflectivity of the collector reflecting surface , γ is the intercept factor, τ is the transmissivity of the glass tube, and α is the absorptivity of the absorber selective coating, Figure 4.1 shows these parameters.

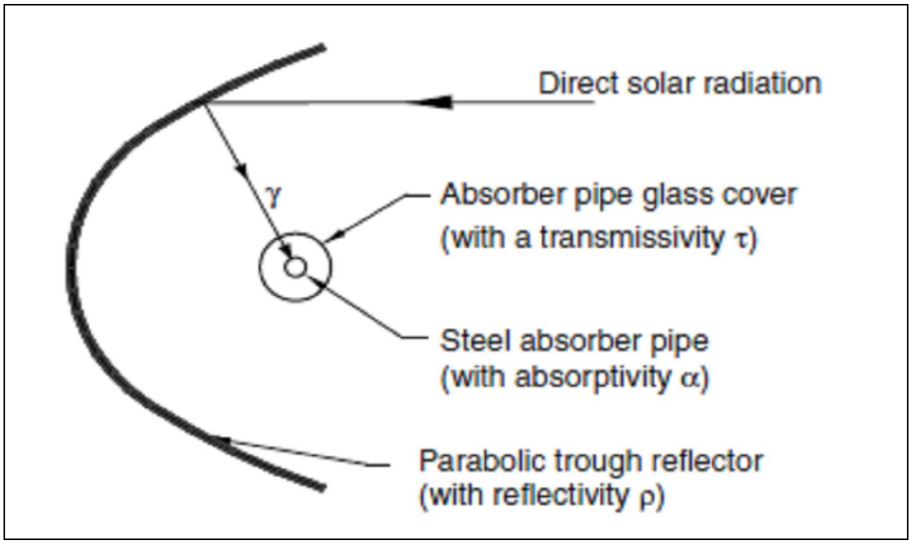


Figure 4.2: Optical parameters of a parabolic trough collector [Duffie J. and Beckman W. (2006)].

The total thermal loss in a PTC, $Q_{collector\ -\ ambient}$, is due to radiative heat loss from the absorber pipe to ambient, $Q_{absorber\ -\ ambient}$, and convective and conductive heat losses from the absorber pipe to its outer glass tube, $Q_{absorber\ -\ glass}$.

Although this heat loss is governed by the well-known mechanisms of radiation, conduction, and convection, it is a good practice to calculate them all together using the thermal loss coefficient, U_l, according to:

$$Q_{collector-ambient} = U_l * \pi * d_o * l * (T_{absmean} - T_{amb})$$

(4.3)

Where T_{amb} is the ambient air temperature, d_0 is the outer diameter of the absorber pipe, and l is the absorber pipe length.

$T_{absmean}$ is the mean absorber pipe temperature and calculated (for the pressurized water and Therminl-VP1) by:

$$T_{absmean} = \frac{T_{in} + T_{out}}{2}$$

(4.4)

Where T_{in} and T_{out} are the inlet and outlet fluid temperatures of the receiver.

The overall heat transfer coefficient U_l is given in W/m^2.K and calculated from the following equation:

$$1/U_l = \frac{1}{h_{in}} + \frac{A_{in}}{A_{out}\, h_{r,c-a}} + \frac{A_{in}}{A_g h_{out}}$$

(4.5)

Where:

A_{in}: Inner receiver area, A_{out}: Outer receiver area, A_g: Glass cover area.

: The convection coefficient between the water and glass cover. h_{in}

: The radiation coefficient between glass cover and ambient. $h_{r,c-a}$
The natural convection coefficient between the receiver and glass cover. h_{out}:
The convection heat transfer coefficient can be calculated by:

$$h_{in} = \frac{N_u K}{L}$$

(4.6)

Where:

The Nusselt Number: $N_u = 0.023 R_e^{0.8} P_r^{0.4}$

(4.7)

The Renolds Number: $R_e = \frac{\rho V D_{out}}{\mu}$

(4.8)

The Prandtl Number: $P_r = \frac{C_p \mu}{K}$

(4.9)

K: the thermal conductivity, L: Receiver length,.

: The fluid density, µ: Fluid dynamic viscosity, V: Fluid velocity. ρ

The radiation coefficient is given by:

$$h_{r,c-a} = \sigma\epsilon(T_{absmean}{}^{4} - T_{amb}{}^{4})/(T_{absmean} - T_{amb})$$

$$=\sigma\epsilon(T_{absmean} + T_{amb})(T_{absmean}^{2} + T_{amb}^{2}) \qquad (4.10)$$

(4.11)

And the natural convection coefficient is calculated by:

$$h_{out} = C(G_r P_r)^n$$

(4.12)

G_r is the Grashof Number and given by:

$$G_r = (gBD_{out}^{3}(T_c - T_{amb}))/v^2$$

(4.13)

Where:

C and n are depened on the geometry of the surface (C = 0.53, n=0.25 for horizontal pipes)[William B. Stine and Michael Geyer. (2001)].

g: Acceleration due to Earth's gravity.

B: Volumetric thermal expansion coefficient.

T_c: Glass cover temperature.

D_{out}: Outer diameter of receiver.

The total energy losses can be calculated from:

Energy losses = Convection losses + Radiation losses

(4.14) $Q_{loss}=Q_{conv.}+Q_{rad.}$

T_{wout} for the pressurized water and Therminol-VP1 oil is calculated from the energy balance equation (using Trial & Error):

(4.15) $$)\dot{m}C_p(T_{out} - T_{in}) = \tau\alpha\rho A_a I_s - U_l A_{in}(T_{absmean} - T_{amb}$$

Substitute U_l and $T_{absmean}$ in equation (4.15) yields:

$$\dot{m}C_p(T_{out}-T_{in}) = \tau\alpha\rho A_a I_s - \left[\frac{1}{h_{in}} + \frac{A_{in}}{A_{out}\left[\sigma\epsilon\left(\frac{T_{in}+T_{out}}{2}+T_{amb}\right)\left(\left(\frac{T_{in}+T_{out}}{2}\right)^2+T_{amb}^2\right)\right]} + \frac{A_{in}}{A_g h_{out}}A_{in}\right]^{-1}\left(\frac{T_{in}+T_{out}}{2} - T_{amb}\right)$$

(4.16)

Where:

: Specific heat of Fluid. C_p

: Direct solar irradiance. I_s

is the mass flow rate of the fluid and can be calculated by: $\dot{m}$

(4.17) $$\dot{m} = (P_{out}/\eta_{th})/(C_p * \Delta T)$$

4.3 Efficiencies of ET collector

The performance of the system is evaluated by the efficiency of the system. There are three different efficiencies used in our simulation:

- Global efficiency (η_g).
- Thermal efficiency (η_{th}).
- Solar efficiency (η_s).

The Global efficiency includes solar and thermal efficiency and can be calculated as:

$$\eta_g = \eta_s * \eta_{th}$$

(4.18)

The thermal efficiency, η_{th}, of the Carnot cycle includes all absorber tube heat losses from conduction, radiation, and convection.

The thermal efficiency is an important measure of performance. It relates parabolic trough collector behavior to the working fluid temperature and assists in predicting overall performance of large-scale systems. Thermal losses occur when energy that has already been absorbed by the receiver is released to the environment. These losses occur by way of heat transfer out of the receiver due to a temperature difference between the receiver and ambient air. The thermal efficiency is given by:

$$\eta_{th} = 1 - \frac{T_{amb}}{T_{wout}}$$

(4.19)

And the solar efficiency is given by:

$$\eta_s = \tau\alpha\rho - \frac{U_l(T_{absmean} - T_{amb})}{CR\ I_s}$$

(4.20)

4.4 Simulation Program

In this thesis the performance of the EuroTrough concentrator as a source of energy for electrical generation will be studied and analyzed. MATLAB software will be used in this study to get the performance of the EuroTrough, and the program will be built to obtain the model conditions, Table 4.1 gives the input and output parameters of the program:

Table 4.1: Input and output Parameters.

Input Parameters	**Output Parameters**
Ambient Temperature (T_{amb})	Absorber Temperature (inlet and outlet)
Water Inlet Temperature (T_{win})	Overall Heat Transfer Coefficient (U_L)
Water Outlet Temperature (T_{wout})	Thermal Loss Collector-Ambient ($Q_{coll-amb}$)

Output Power Per Collector (P_o)	Thermal Efficiency (η_{th})
Specific Heat of Water (C_{pw})	Global Efficiency (η_g)
Outer diameter of steel absorber pipe (d_o)	Solar Efficiency (η_s)
Gross length of every concentrator module (L)	Water Mass Flow Rate (m_w)
Direct Solar Irradiance (I_s)	Output Power (P_o)
Parabola width (L_a)	Aperture Area (A_{apr})

The flow chart of MATLAB program is shown in figure 4.2, it starts by calculating $T_{absmean}$ and U_l as function of T_{out}, the solution procedure assumes different values of direct solar irradiance I_s. The main step of the program is to estimate the fluid outlet temperature by Trial & Error method at each direct solar irradiance step. Then the mass flow rate $\dot{m}$, the total thermal loss $Q_{coll\text{-}amb}$, $T_{absmean}$, overall heat transfer coefficient U_l and thermal efficiency are calculated based on the the values of T_{out}. The solar efficiency is obtained based on $T_{absmean}$ and U_l. Finally the global efficiency is calculating from the solar and thermal efficiencies.

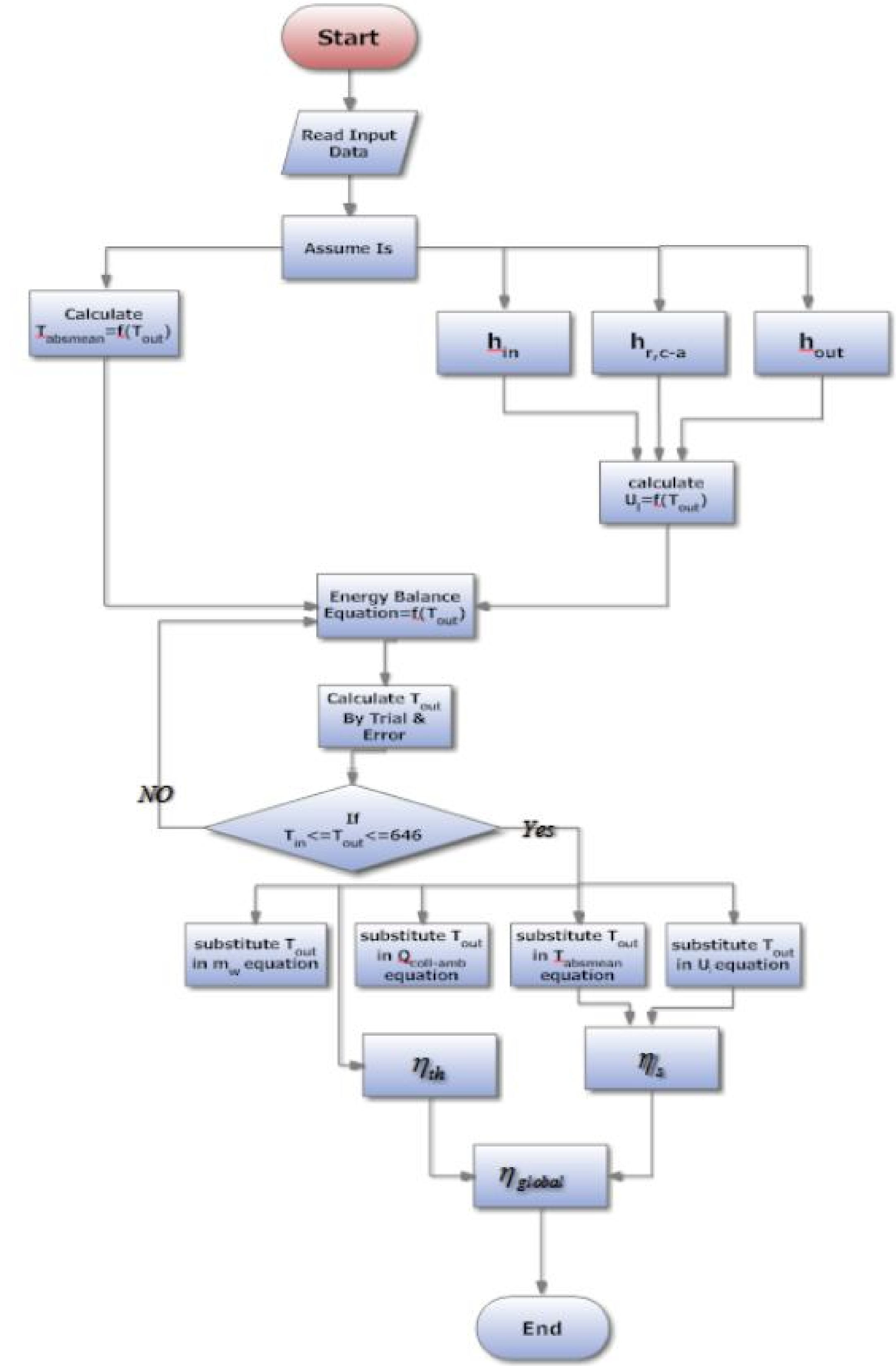


Figure 4.2: Flow chart of simulation program.

Chapter Five: Results and discussion.

5.1 Introduction

As mentioned previously, the HTF in ET collector will be replaced firstly by pressurized water and then boiling water, so two main cases will be simulated with different case for each of them. In the previous chapter, Chapter Four, the input and output parameters of simulation model are mentioned in Table 4-1. In this chapter, different cases will be studied and carried out, also a comparison with simulation results with previous results will be done.

5.2 Pressurized Water

The pressurized water is water under high pressure which maintains it liquid when the temperature increases.

A MATLAB code is written based on the mathematical model of collector which described in chapter three (see appendix A). The input collector parameters used in case of variation the direct solar irradiance (I_s) are shown in Table 5-1

Table 5-1 input collector parameters with varied Is

T_{amb} = 30+273 = 303°K	**T_{win}= 50+273 =323°K**	**P_{out}= (200/8) = 25 KW/module**	**C_{pw} = 4.38 J/g.K**
l=12.27 m	**l_a = 5.76 m**	**A_{apr} = l_a*l = 70.6752 m^2**	**d_o = 0.07 m**

Depending in Table 5-1 and assume I_s (it can be calculated from equation 5.5) varied from 0.2 to 1 with step 0.1 kW/m^2, the performance results when we run the program is shown in Table 5-2.

Table 5-2 Output Performance of ET Collector with varied Is

T_{wout} (°K)	$T_{abs\text{-}mean}$ (°K)	U_L (W/m²K)	$Q_{collector\text{-}amb.}$ (W)	m_w (kg/s)	$\eta_{thermal}$ (%)	η_s (%)	η_g (%)	I_s (kW/m²)
581.21	452.10	2.063	829.71	0.0462	47.87	52.89	25.32	0.2
589.35	456.17	2.395	989.37	0.0441	48.59	58.89	28.61	0.3
595.97	459.48	2.745	1158.4	0.0425	49.16	61.73	30.35	0.4
601.4	462.20	3.109	1335.0	0.0413	49.62	63.32	31.42	0.5
605.89	464.44	3.485	1517.5	0.0404	49.99	64.32	32.15	0.6
609.63	466.31	3.869	1704.0	0.0396	50.3	64.99	32.69	0.7
612.75	467.87	4.257	1892.9	0.039	50.55	65.47	33.1	0.8
615.37	469.18	4.646	2082.3	0.0385	50.76	65.84	33.42	0.9
617.58	470.29	5.033	2270.8	0.038	50.94	66.14	33.69	1.0

As shown in Table 5-2, changing I_s gives direct relationship with efficiencies. Figure 5-1 shows graphically the relationship between the direct solar irradiance and the solar, global and thermal efficiencies.

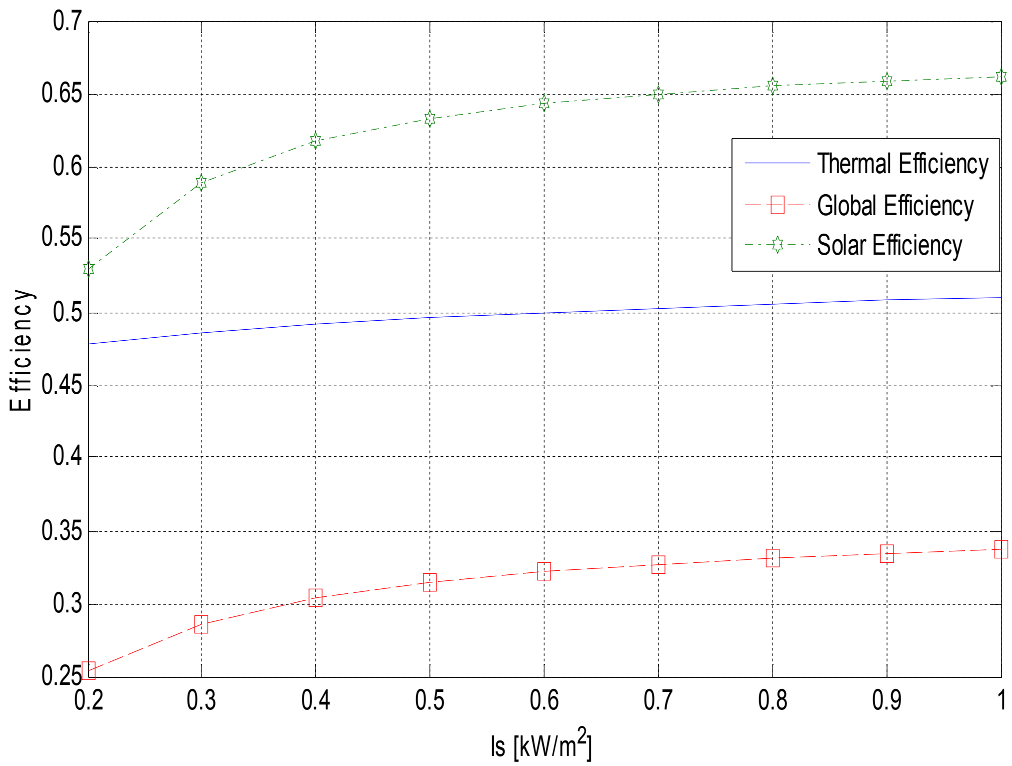


Figure 5.1 EuroTrough performance due to variation of direct solar irradiance

Similarly Figure 5-2 shows the overall heat transfer coefficient versus the water output temperature, while Figure 5-3 demonstrates the relationship between the water output temperature and thermal loss collector-ambient.

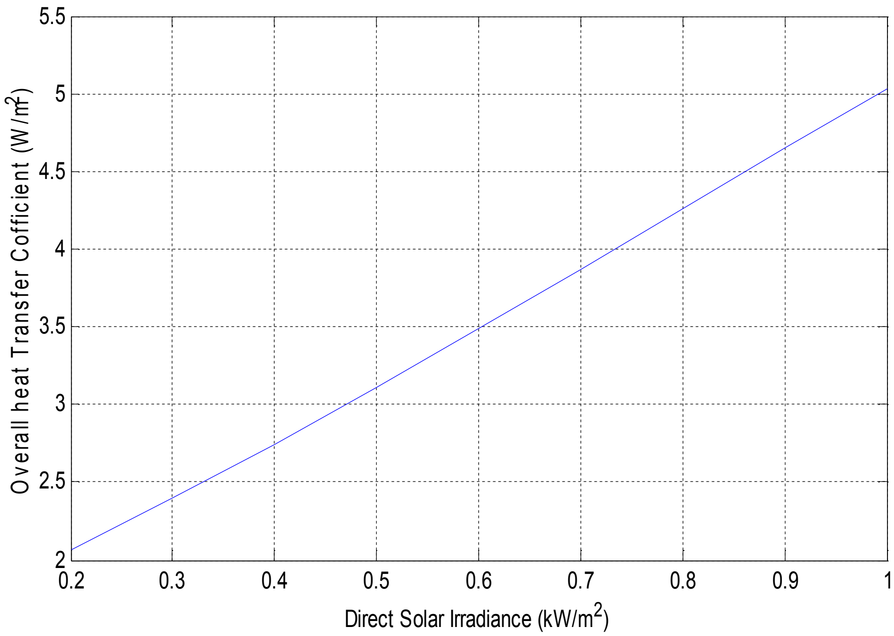


Figure 5.2 Overall heat transfer coefficient with water output temperature.

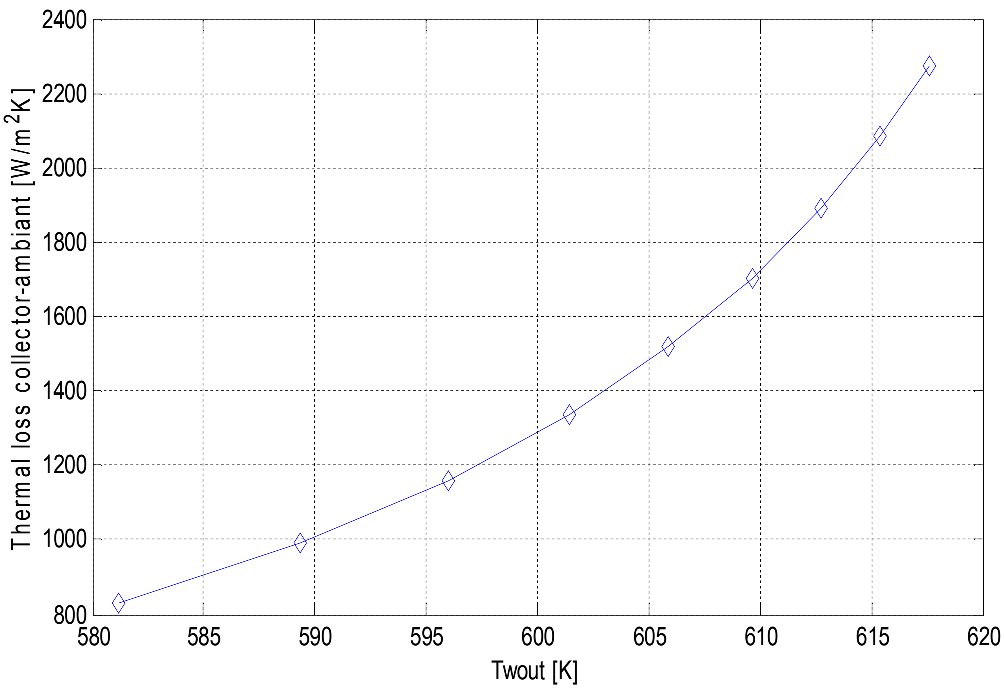


Figure 5.3 Thermal losses collector-ambient versus water output temperature

5.3 Boiling Water

As shown in Figure 5-4, the temperature of the receiver does not increase linearly with receiver length as a result of the phase change; the average temperature of the receiver is not simply the average of the receiver inlet and outlet fluid temperatures. A reasonable approach to computing the average receiver temperature is to first calculate the average temperature of the receiver preheat, boiling, and super heat regions individually. These average temperatures, weighted by the length of the respective receiver regions, are then used to compute an overall receiver average temperature. [William B. Stine and Michael Geyer. (2001)]

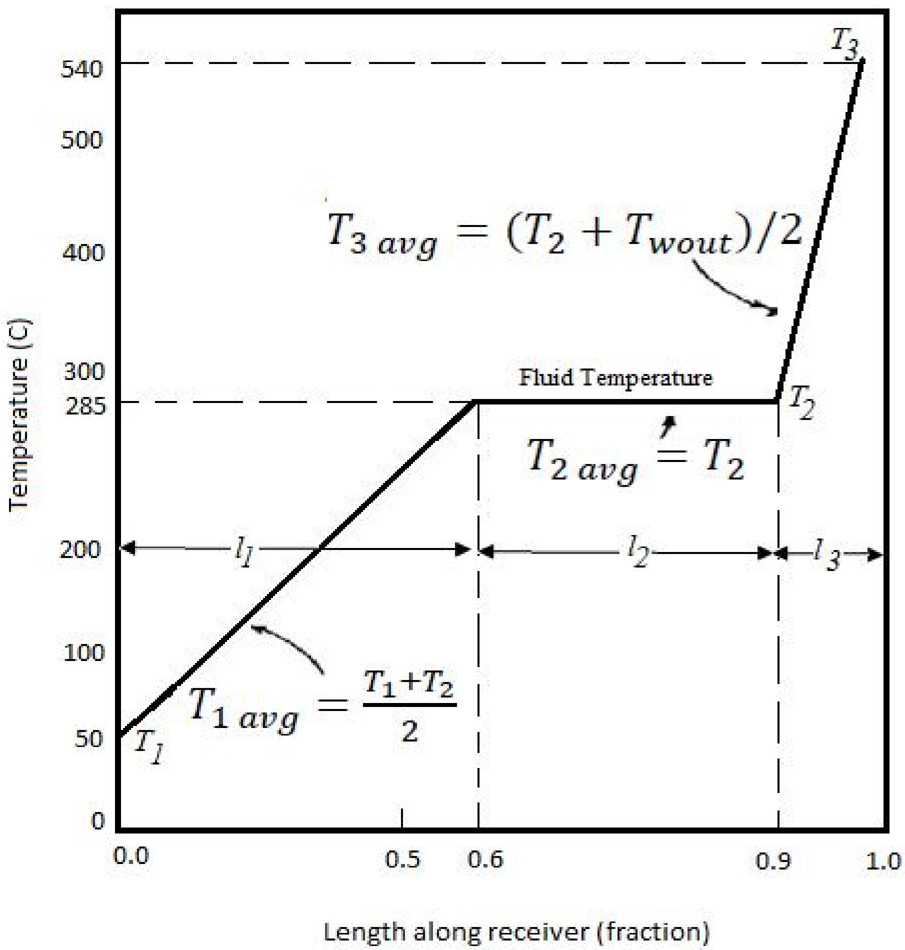


Figure 5-4 Temperature distribution along the receiver of the PTC at 6.2 MPa [William B. Stine and Michael Geyer**. (2001)**].

Depending on figure 5-4, the average temperatures along the receiver were calculated as:

$$T_{1\,avg} = \frac{T_1+T_2}{2},\; T_{2\,avg} = T_2 \;\; and\, T_{3\,avg} = (T_2 + T_{wout})/2$$

(5.1)

Overall average temperature of receiver is given by:

$$T_{absmeanboil.} = T_{1\,avg}\frac{l_1}{L} + T_{2\,avg}\frac{l_2}{L} + T_{3\,avg}\frac{l_3}{L}$$

(5.2)

Where the total length of the receiver is:

$$L = l_1 + l_2 + l_3$$

(5.3)

Using steam tables to obtain the values of enthalpy at 6.2 MPa, and substitute them and $T_{absmeanboil.}$ (from equation (5.2)) in the energy balance equation to find outlet temperature of the boiling water:

$$\dot{m}_w(h_{out} - h_{in}) = \tau\rho\alpha A_a I_s - \left[\frac{1}{h_{in}} + \frac{A_{in}}{A_{out}\left[\sigma\epsilon\left(\frac{T_{in}+T_{out}}{2}+T_{amb}\right)\left(\left(\frac{T_{in}+T_{out}}{2}\right)^2+T_{amb}^2\right)\right]} + \frac{A_{in}}{A_g h_{out}}A_{in}\right]^{-1}\left[A_{in}\left(T_{1\,avg}\frac{l_1}{L} + T_{2\,avg}\frac{l_2}{L} + \left(\frac{T_2+T_{wout}}{2}\right)\frac{l_3}{L} - T_a\right)\right]$$

(5.4)

Where h_{out} and h_{in} are the enthalpy at the inlet and outlet water temperatures of the receiver.

Referring to MATLAB code shown in appendix A, the results when the boiling water acts as working fluid with variable direct solar irradiance in range of 0.2 to 1.0 kW/m^2 and with h_f=209.26 kJ/kg at 323^{o}K are given in Table 5-3.

Table 5-3 Output Performance of ET Collector with boiling water and varied Is

T_{wout} (^{o}K)	$T_{abs\text{-}mean}$ (^{o}K)	U_L (W/m^{2}K)	$Q_{collector\text{-}amb.}$ (kW)	m_w (Kg/s)	$\eta_{thermal}$ (%)	η_s (%)	η_g (%)	h_{out} kJ/kg	I_s (kW/m^2)

560	441.5	5.0723	1.89	0.0550	45.89	15.32	7.03	2797.0	0.2
568	445.5	5.0959	1.94	0.0523	46.65	36.31	16.94	2845.0	0.3
573	448	5.1107	1.98	0.0508	47.12	47.17	22.23	2875.0	0.4
583	453	5.1401	2.05	0.0479	48.02	53.21	25.55	2935.4	0.5
593	458	5.1693	2.12	0.0453	48.91	57.23	27.99	2962.1	0.6
606	464.5	5.2071	2.21	0.0423	50.00	59.89	29.95	3001.7	0.7
618	470.5	5.2417	2.29	0.0398	50.97	61.95	31.58	3033.7	0.8
633	478	5.2847	2.39	0.0370	52.13	63.4	33.05	3075.0	0.9
640	481.5	5.3046	2.44	0.0358	52.65	64.93	34.19	3075.2	1.0

According to Table 5.3, the relationship between solar, thermal and global efficiencies and direct solar irradiance can be plotted as shown in Figure 5-5.

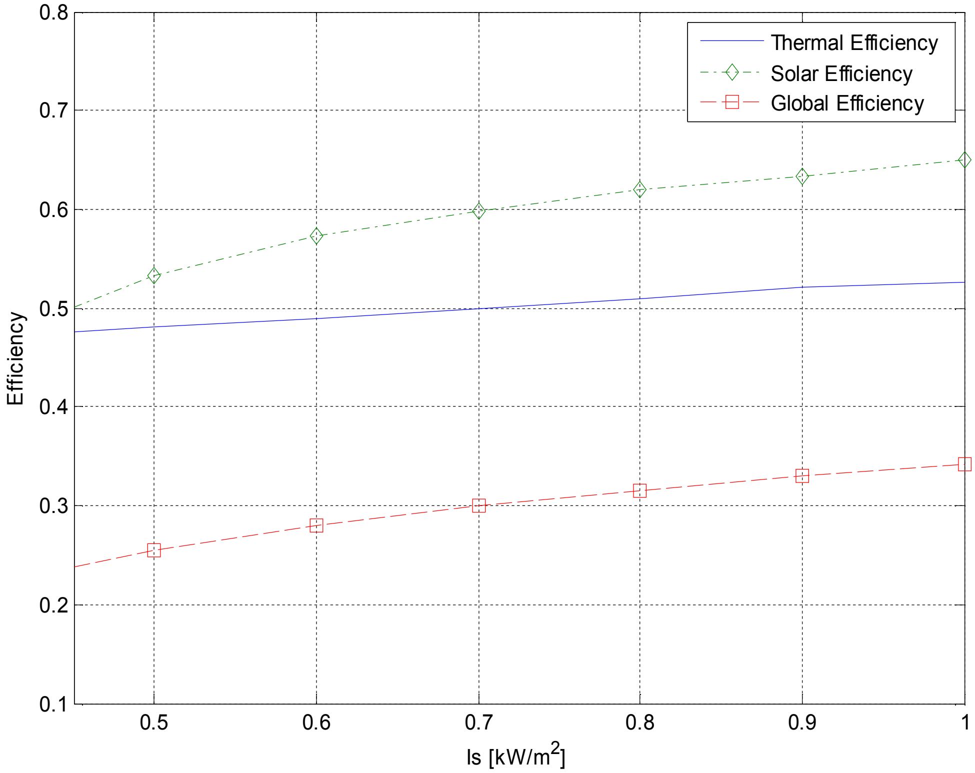


Figure 5-5: Performance of ET collector when boiling water is used in the receiver

Also the overall heat transfer coefficient versus the water output temperature relationship is obtained as shown in Figure 5-6, while Figure 5-7 demonstrates the relationship between the water output temperature and thermal loss collector-ambient.

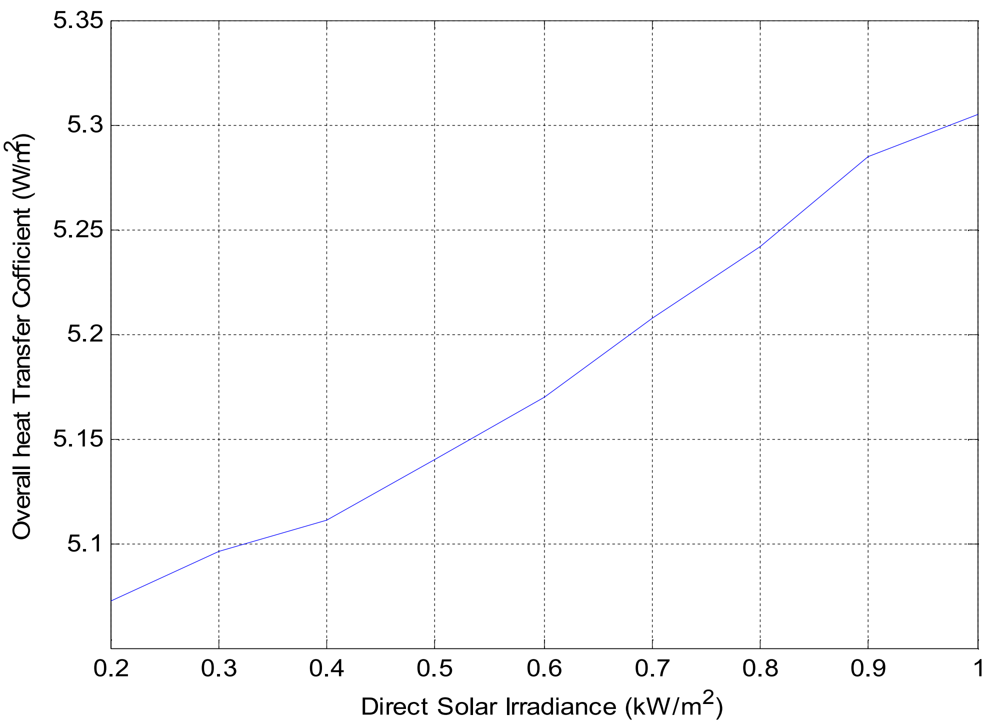


Figure 5.6 Overall heat transfer coefficient with boiling water output temperature

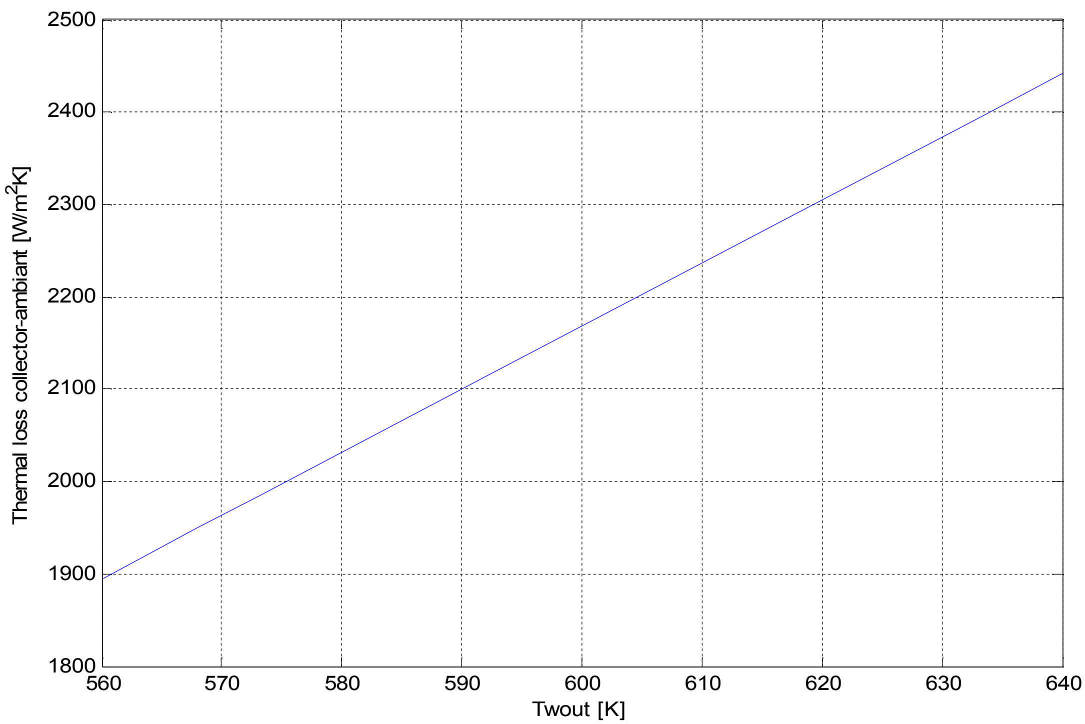


Figure 5.7 Thermal loss collector-ambient with boiling water output temperature

5.4 **Therminol VP1 thermal oil as working fluid**

Therminol VP-1 heat transfer fluid is a synthetic heat transfer fluid designed to meet the demanding requirements of vapor phase systems from 257° C - 400° C or liquid phase systems from 12° C - 400° C. Some of the thermal physical properties for hot oil for an average value of the operation temperature are given in Table 5-4.

Table 5-4 Properties of the **Therminol-VP1 oil [Kreith K and Goswami Y. (2007)]**

	Properties of the fluid
c_p specific heat at constant pressure (J/kg K)	2436.65 J/(kg·K)
P density of the fluid	764.63 kg/m^3
M Dynamic viscosity	165.54·10^{-6} (kg/m·s)

In this thesis, Therminol-VP1 thermal oil is used and tested also as working fluid. The MATLAB code was written to calculate the performance of ET collector used oil as working fluid is mentioned in appendix A. Changing direct solar irradiance in range of 0.2 to 1.0 kW/m^{2,} the behavior of the solar and global efficiencies are shown in Table 5-5. Figure 5-8 shows the performance of ET when Therminol-VP1 thermal oil is used as working fluid.

Table 5-5 Results of ET collector when used oil as working fluid.

T_{wout} (°K)	$T_{abs-mean}$ (°K)	U_L (W/m^{2}K)	$Q_{collector-amb.}$ (kW)	m_w (Kg/s)	$\eta_{thermal}$ (%)	η_s (%)	η_g (%)	I_s (kW/m^2)
522.1	422.52	1201.9	1.20	0.152	41.96	39.76	16.68	0.2

536.3	429.66	1456.0	1.45	0.136	43.51	47.92	20.85	0.3
547.9	435.47	1717.4	1.71	0.126	44.7	51.87	23.19	0.4
557.5	440.22	1982.6	1.98	0.118	45.65	54.19	24.73	0.5
565.3	444.14	2248.1	2.24	0.113	46.4	55.73	25.86	0.6
571.8	447.40	2510.8	2.51	0.108	47.01	56.86	26.73	0.7
577.2	450.11	2768.3	2.76	0.105	47.51	57.75	27.44	0.8
581.8	452.39	3018.4	3.01	0.102	47.92	58.5	28.03	0.9
585.6	454.32	3259.3	3.25	0.100	48.26	59.17	28.56	1.0

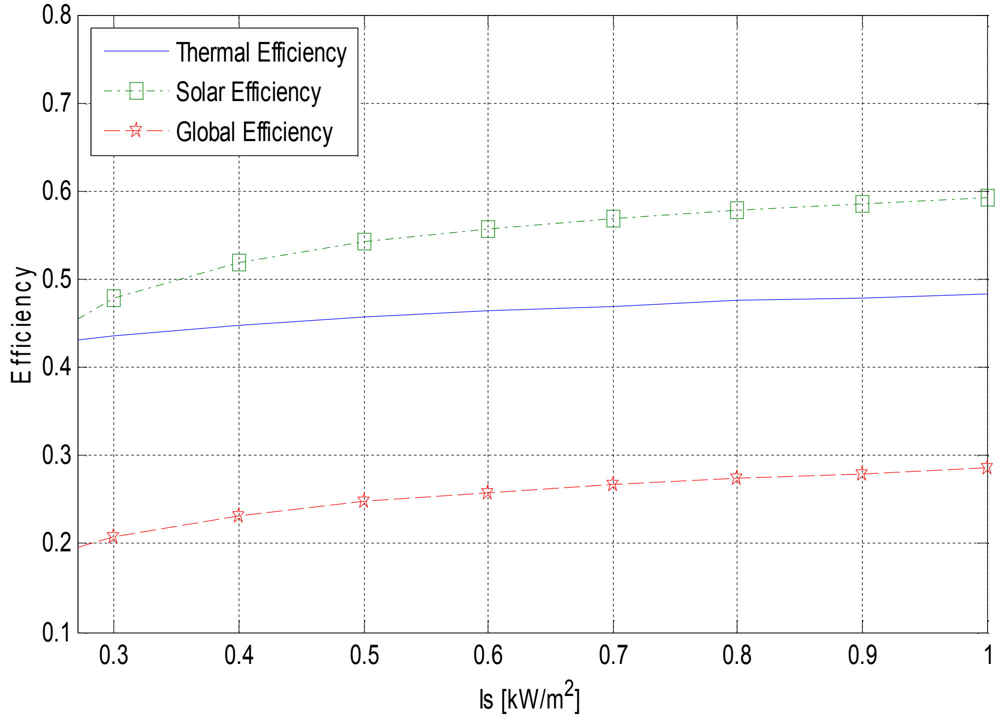


Figure 5-8 Performance of ET when used oil as working fluid.

The overall heat transfer coefficient and thermal loss collector-ambient results when Therminol- VP1oil is used as working fluid are shown in Figure 5-9 and Figure 5-10 respectively.

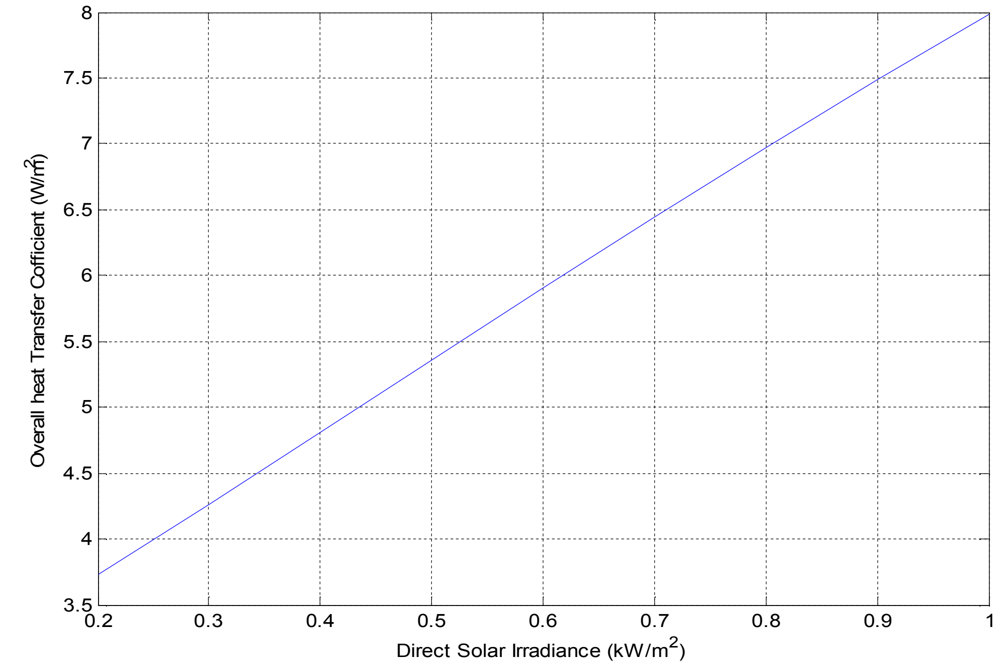


Figure 5-9: The overall heat transfer coefficient when Therminol-VP1 oil is used

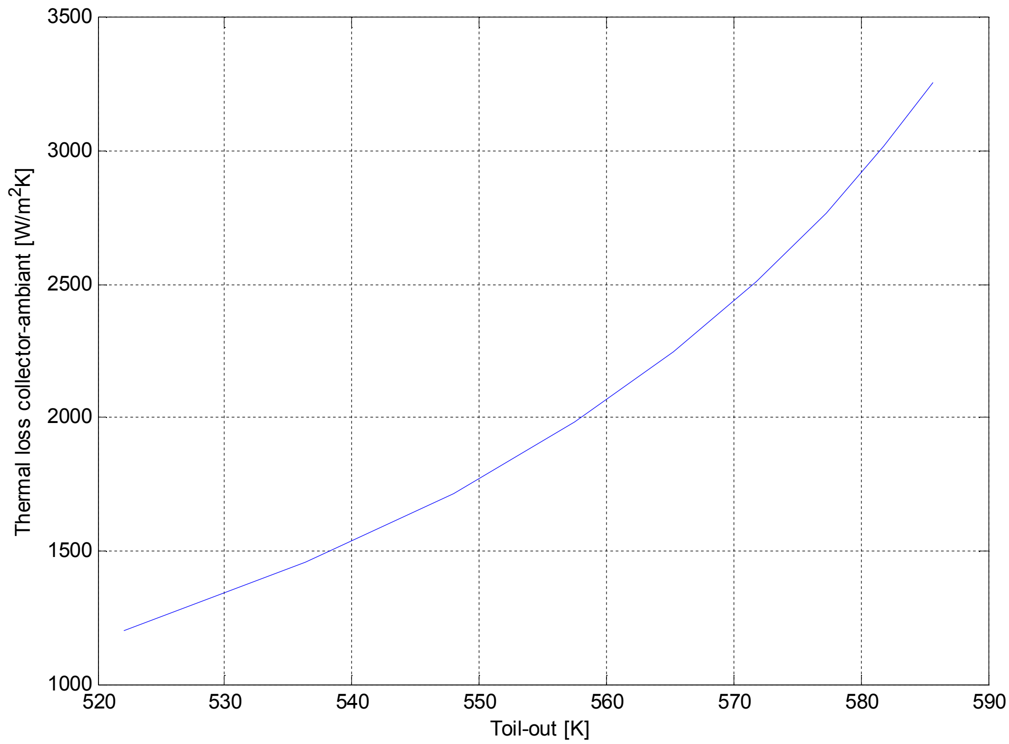


Figure 5-10: The thermal loss collector-ambient when Therminol-VP1 oil is used

5.5 Application of ETC in Jordan

Unlike other countries of the Middle East, Jordan is a non-oil producing country. Its domestic recoverable energy resources are limited and don't satisfy the demands of increasing population and economic growth. Nearly all the generated electricity in Jordan is produced from power plants that use fossil fuels. The most popular fuels are heavy fuel oil and diesel fuels. However, this option is not very attractive since Jordan's spending on petroleum is more than 50% of its export earnings. Therefore, switching to renewable energy sources for electricity generation is important since it serves as an optimal solution for energy and environmental issues.

The Direct Solar Irradiance I_s (the rate at which radiant energy is incident on a surface per unit area) is different from a place to another, I_s for a horizontal surface at any time between sunrise and sunset is given by [Duffie J. and Beckman W. (2006)]:

$$I_s = G_{sc} * K_t \left[1 + 0.033 * \cos\left(\frac{360 * n}{365}\right)\right] * (\sin\phi \sin\delta + \cos\phi \cos\delta \cos\omega)$$

(5.5)

Where δ is the Declination and can be calculated by [Duffie J. and Beckman W. (2006)]:

$$\delta = 23.45 * \sin\left(360 * \frac{284 + n}{365}\right)$$

(5.6)

Where:

: The solar radiation. I_s

: Solar constant = 1367 W/m^2 G_{sc}

K_t : The clearness index= 0.8.

: Latitude. ϕ

: The hour angle. ω

: The day of the year; n can be conveniently obtained with the help of Table 5-6 n [Duffie J. and Beckman W. (2006)].

Table 5-6 The day of the year for each month.

		For the average Day of the Month		
Month	N for ith Day of Month	Date	N, Day of year	δ, Declination
January	I	17	17	-20.9
February	31+i	16	47	-13.0
March	59+i	16	75	-2.4
April	90+i	15	105	9.4
May	120+i	15	135	18.8
June	151+i	11	162	23.1
July	181+i	17	198	21.2
August	212+i	16	228	13.5
September	243+i	15	258	2.2
October	273+i	15	288	-9.6
November	304+i	14	318	-18.9
December	334+i	10	344	-23.0

Applying our simulation program to calculate the Solar irradiance, Solar Efficiency, Global Efficiency and Thermal Efficiency for some places in Jordan, by using equation 5-5 to calculate the direct solar irradiance for each place, and by taking two days of the year at the same time 3:00 pm (ω), we can get these results in Table 5-7.

Table 5-7 ET results applied in some places in Jordan.

Irbid (ϕ=32.55)	Direct Solar Irradiance (kW/m^2)		Solar Efficiency (%)		Global Efficiency (%)		Thermal Efficiency (%)	
	16/march (n=75)	16/August (n=228)	16/march (n=75)	16/August (n=228)	16/march (n=75)	16/August (n=228)	16/march (n=75)	16/August (n=228)
	0.626	0.745	61.30	64.71	30.66	32.30	49.88	49.98
Aqaba (ϕ=29.52)	0.649	0.754	62.06	64.91	31.02	32.40	49.99	50.01
Amman (ϕ=31.57)	0.633	0.748	61.67	64.78	31.01	32.33	49.88	49.91

Similarly applying our simulation program to calculate the Solar irradiance, Solar Efficiency, Global Efficiency and Thermal Efficiency for Ma'an city for the all months around the year as shown in Table 5-8 (for pressurized water), and Table 5-9 (for Therminol-VP1 oil), Figures 5-11 and 5-12 show these results.

Table 5-8 ET results applied in Ma'an when pressurized water is used.

Month	Date	N, Day of year	Solar Irradiance (kW/m^2)	Global Efficiency (%)	Solar Efficiency (%)	Thermal Efficiency (%)

January	17	17	0.437	26.15	52.14	50.16
February	16	47	0.535	28.89	57.69	50.18
March	16	75	0.644	30.95	61.91	50.2
April	15	105	0.736	32.2	64.49	50.22
May	15	135	0.784	32.72	65.59	50.25
June	11	162	0.797	32.85	65.87	50.29
July	17	198	0.788	32.76	65.68	50.08
August	16	228	0.752	32.38	64.88	50.0
September	15	258	0.676	31.43	62.89	49.93
October	15	288	0.566	29.57	59.07	49.89
November	14	318	0.46	26.89	53.64	49.88
December	10	344	0.408	25.07	49.96	49.87

Table 5-9 ET results applied in Ma'an when Therminol-VP1 is used.

Month	Date	N, Day of year	Solar Irradiance (kW/m^2)	Global Efficiency (%)	Solar Efficiency (%)	Thermal Efficiency (%)
January	17	17	0.437	22.04	48.29	45.65
February	16	47	0.535	24.86	54.54	45.66
March	16	75	0.644	26.98	59.29	45.69
April	15	105	0.736	28.27	62.2	45.72
May	15	135	0.784	28.81	63.43	45.75
June	11	162	0.797	28.95	63.75	45.79
July	17	198	0.788	28.85	63.53	45.71
August	16	228	0.752	28.46	62.63	45.65
September	15	258	0.676	27.47	60.4	45.62
October	15	288	0.566	25.56	56.09	45.59
November	14	318	0.46	22.81	49.97	45.54
December	10	344	0.408	20.93	45.84	45.51

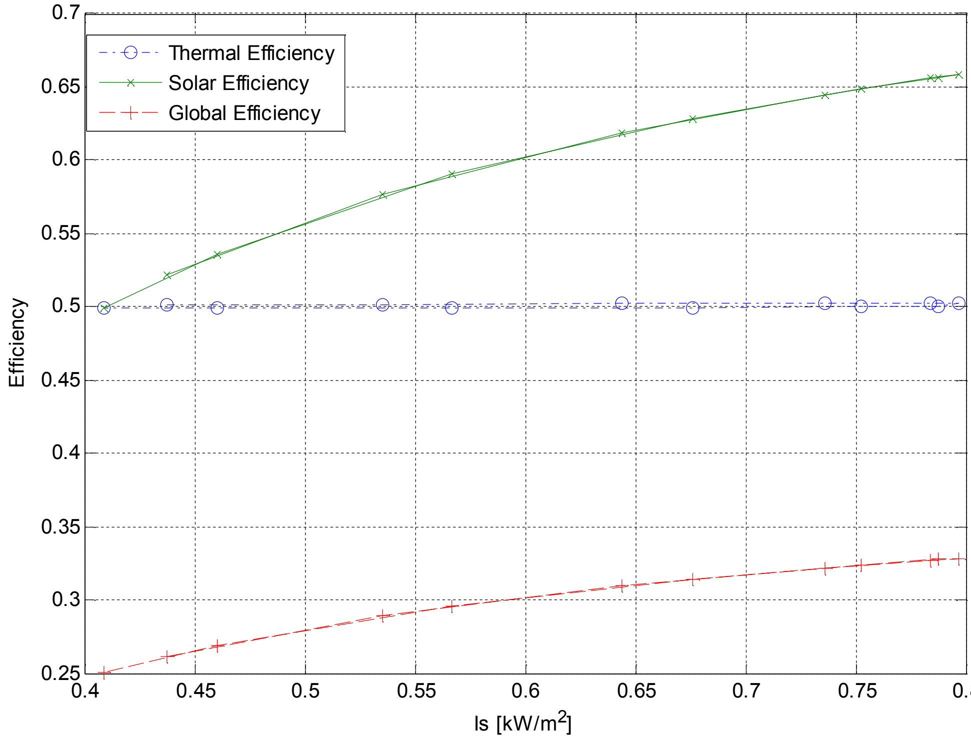


Figure 5-11 Performance of ET applied in Ma'an when pressurized water is used.

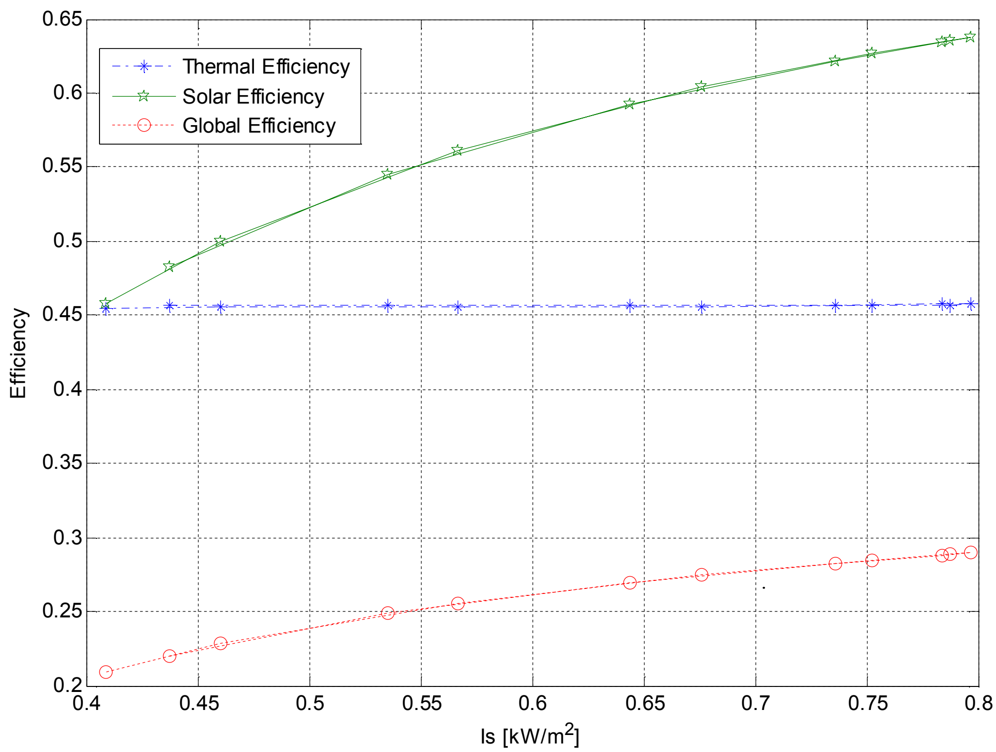


Figure 5-12 Performance of ET applied in Ma'an when Therminol-VP1 is used.

Finally the Direct solar irradiance distribution in Ma'an along one year was calculated as shown in table 5-8and plotted in Figure 5-13.

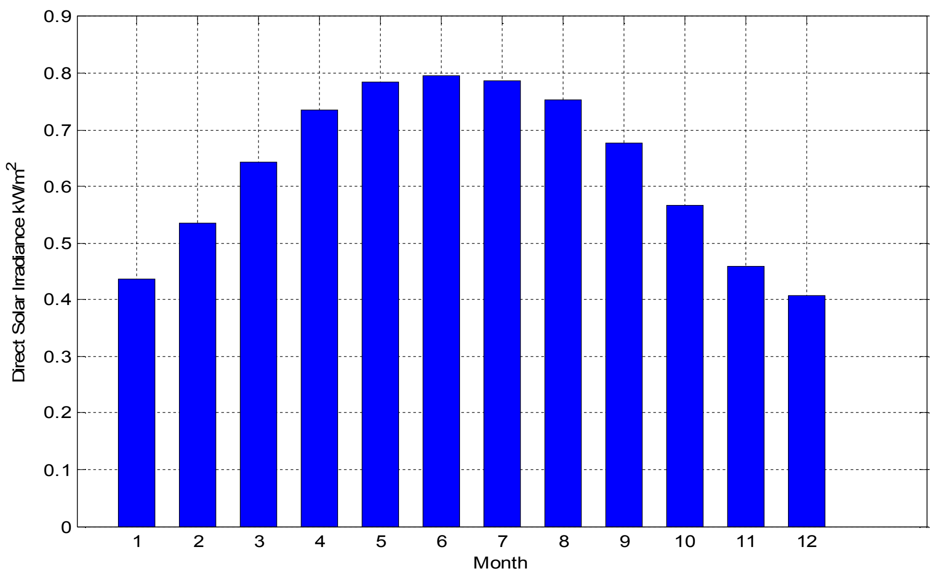


Figure 5-13 Direct solar irradiance on horizontal plane in Ma'an

5.6 Validation

Efficiency is an important measure of performance. It relates parabolic trough solar collector behavior to the working fluid temperature and assists in predicting overall performance of systems. In this section a comparison between different cases studies in euro-trough collector will be worked. Also other results were founded in literature will be compared to this work.

5.6.1 Comparison between the simulated results and those from literature

Data from thermal efficiency simulation are consolidated in Figure 5.14. The best fit curve in this Figure was obtained from a linear analysis equation with a negative slope of -0.049, which yields the following equation:

$$\eta_{th} = -0.049\frac{\Delta T}{I_s} + 51.85$$

(5-4)

According to Figure 5.14, the point at which the performance curves corresponds to a thermal efficiency of 51.84% is ($\Delta t/I_s$) = 0.2 m^2 K/W.

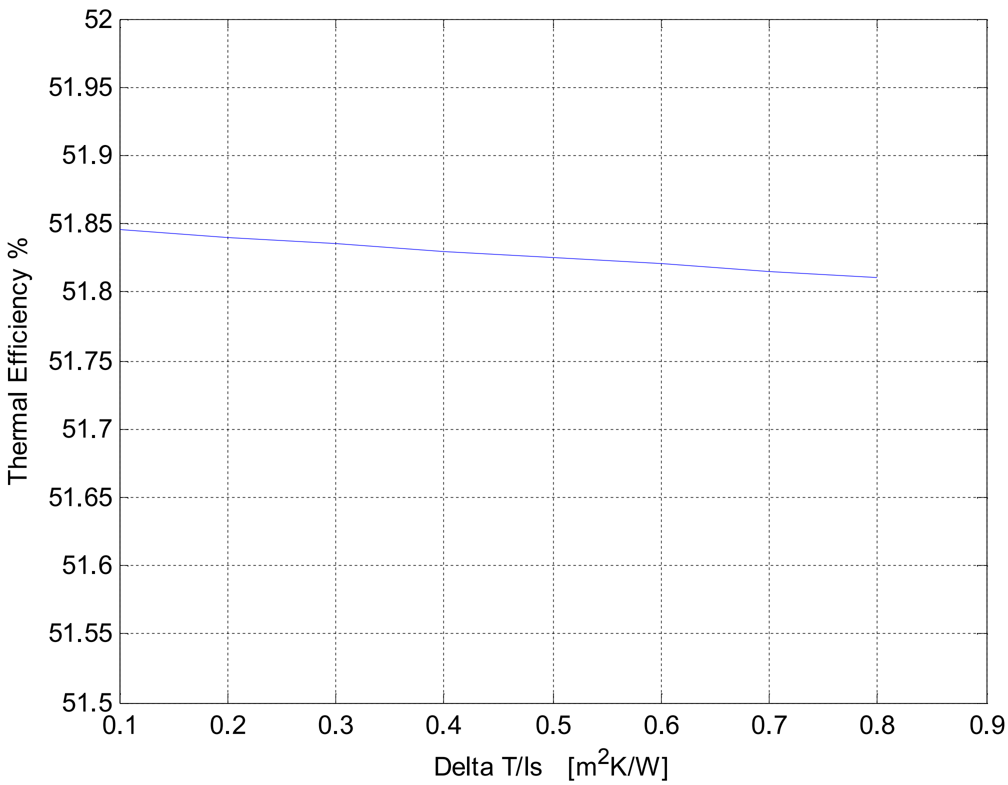


Figure 5-14 Thermal Efficiency versus $\Delta T/I_s$.

In literature, (Brooks M. and Harms T) obtain thermal efficiency versus ($\Delta t/I_s$) as in Figure 5-15, where at ($\Delta t/I_s$) = 0.2 m^2 K/W the thermal efficiency was 52 %, and 51.5% for shielded class and unshielded class respectively.

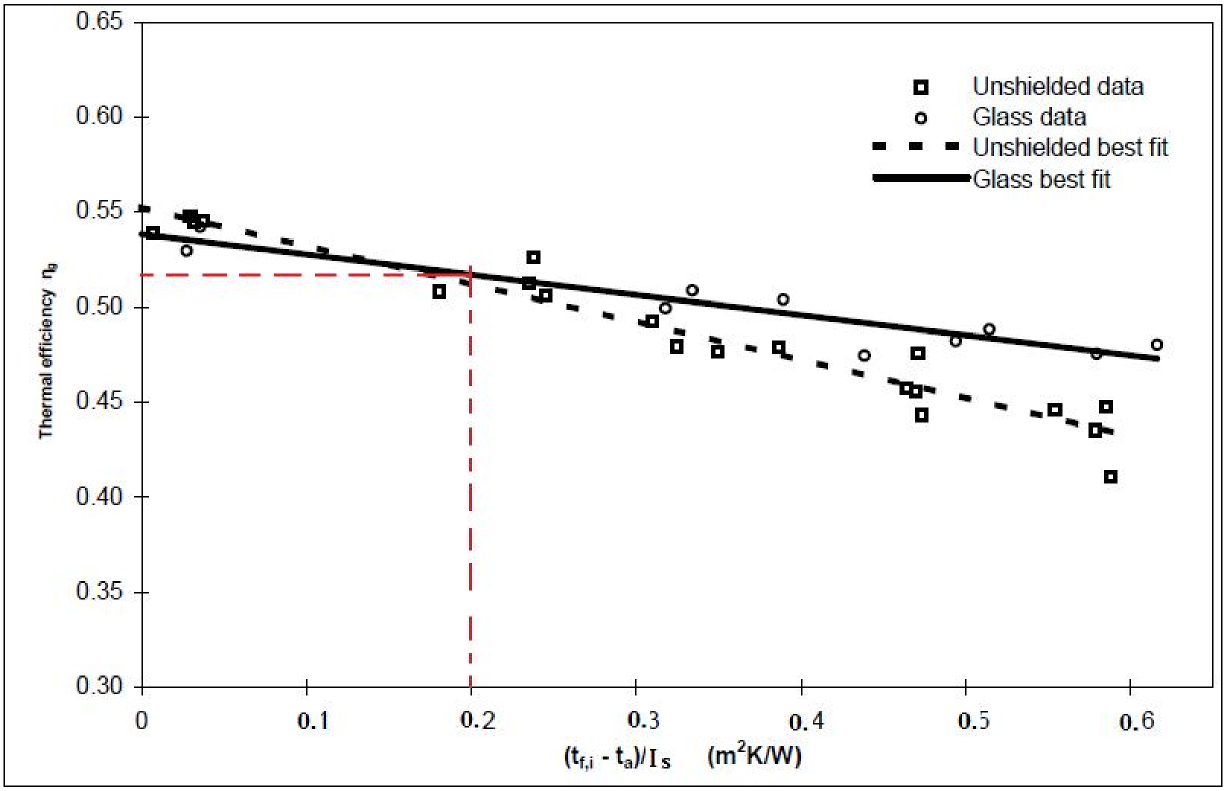


Figure 5-15 Comparison of thermal efficiency [Brooks M. Mills I. and Harms T. (2006)].

Other case, we will compare results of solar efficiency and ($\Delta t/I_s$) curves with those results were founded in literature. Figure 5-16 shows the curve between $(T_{win} - T_{amb})/I_s$ and the solar efficiency of EuroTrough which we obtain in this simulation. Similarly as thermal versus ($\Delta t/I_s$) curves, using linear analysis we have the following equation

$$\eta_s = -0.156\frac{\Delta T}{I_s} + 69.29$$

(5-5)

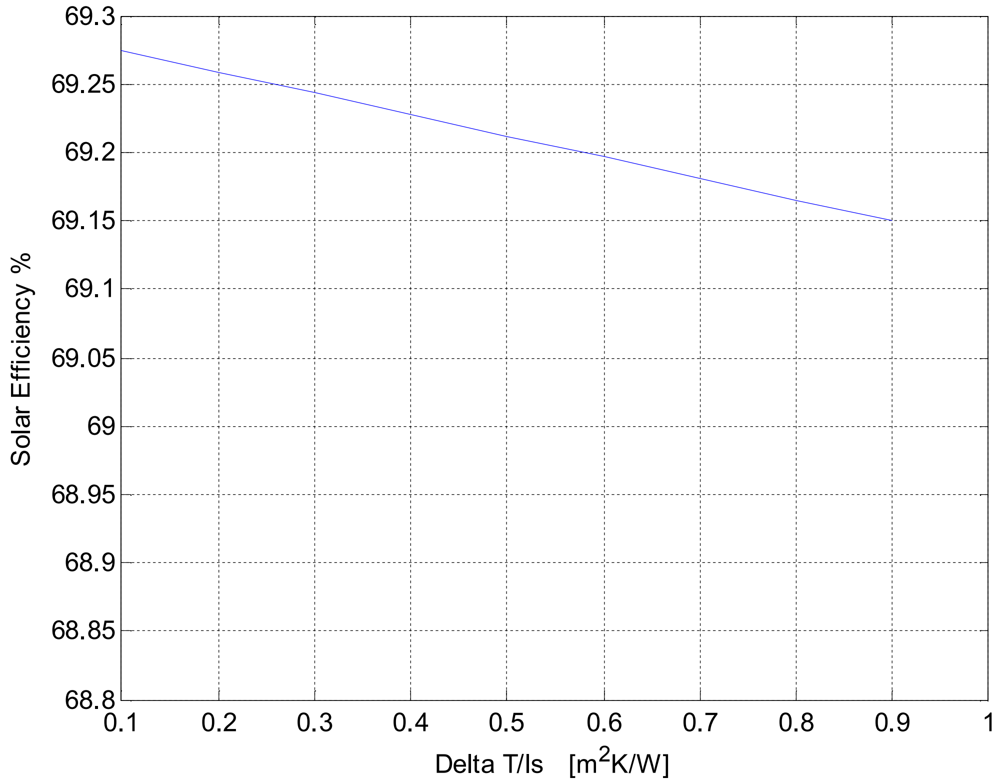


Figure 5-16: solar efficiency versus $\Delta T/I_s$ for ET collector.

By taking the point of $(T_{win} - T_{amb})/I_s = 0.3\ m^2K/W$, the solar efficiency $\eta_s = 69.24\%$, comparing this result with LS-2 collector (Bialobrzeski R. (2007) as shown in Figure 5-17, we found that at the same value of $(T_{win} - T_{amb})/I_s$ the solar efficiency is 50% which is smaller than the value that obtained for ET from our simulation.

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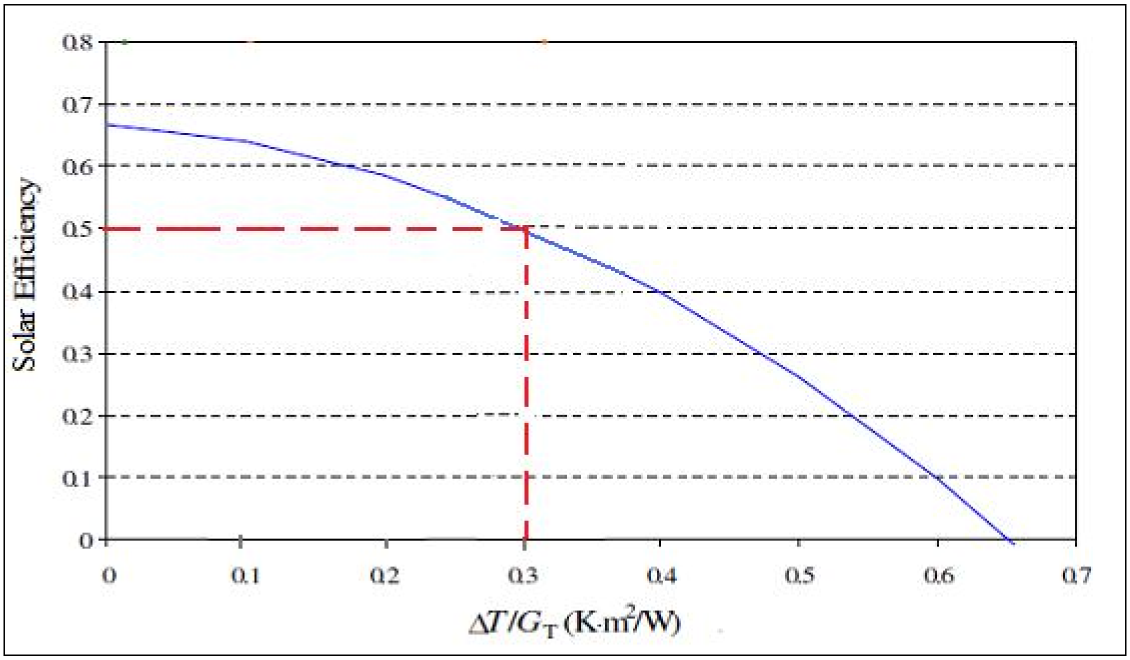


Figure 5-17: ΔT/I_s versus solar efficiency for LS-2 parabolic trough collector.[Bialobrzeski R. (2007)].

5.6.2 Comparison of efficiencies for the ET with different working fluids

Two working fluids were used in the simulation which are: water (Boiling & Pressurized) and oil (Therminol-VP1).

Figure 5-18 compares between solar efficiency for different working fluid in the receiver tube of euro-trough collector. As shown, the solar efficiency with pressurized water acts as working fluid has the highest efficiency compared to other while the oil has the lowest solar efficiency. Similarly, Figure 5-19 and Figure 5-20 show the global efficiency and thermal efficiency of ET collector, where the water (Boiling & Pressurized) has the highest values and the oil has the lowest results.

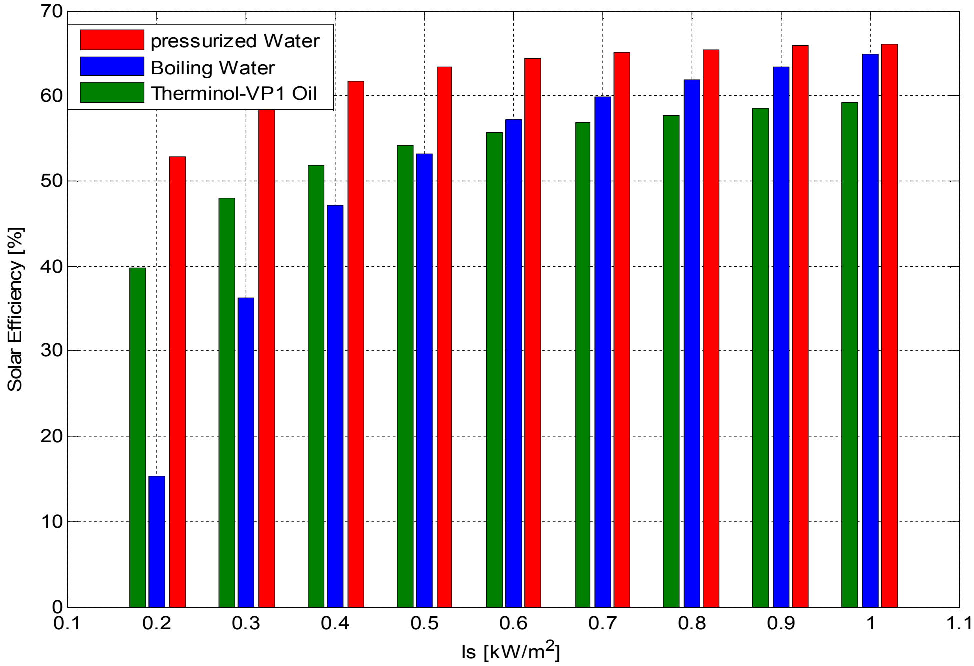


Figure 5-18 Comparison of solar efficiency with different working fluids.

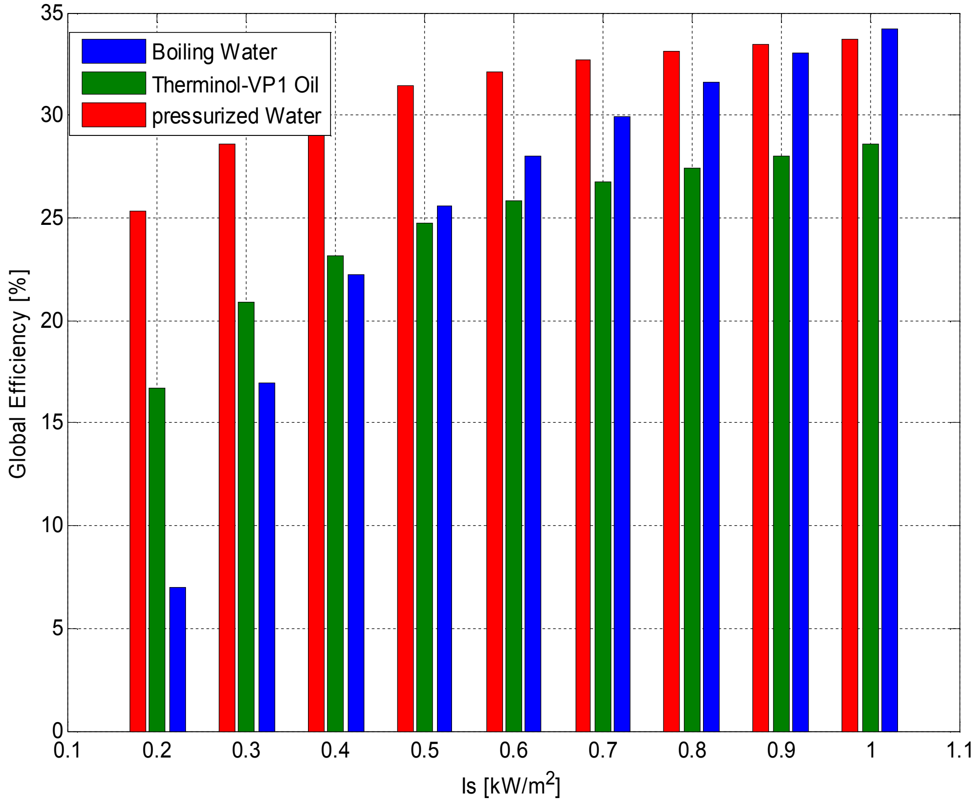


Figure 5-19 Comparison of global efficiency with different working fluids.

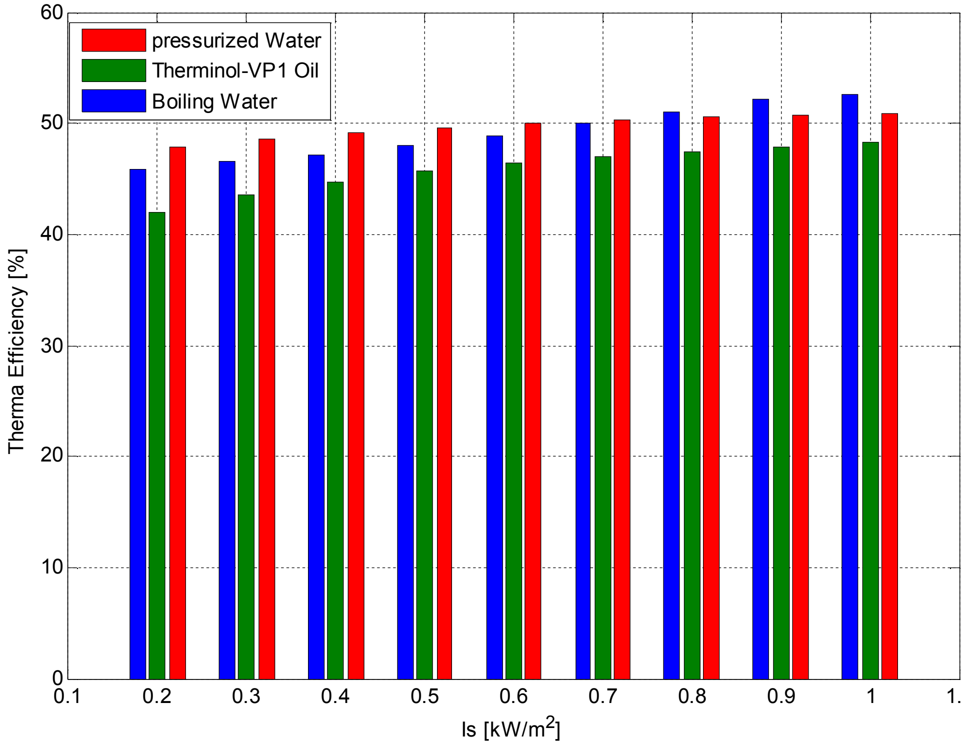


Figure 5-20 Comparison of thermal efficiency with different working fluids.

Chapter Six: Conclusion and Recommendations

6.1 Conclusion

This thesis deals with the performance of a solar parabolic trough (EuroTrough collector), as a source of energy for electrical generation.

The purpose of this thesis is to study and analyze the performance of EuroTrough collector by using water (Pressurized and Boiling) and Therminol-VP1 oil in the receiver tube as a working fluid. We can conclude from the simulation and calculations shown in the previous chapters that using water (either pressurized or boiling) in the receiver tube is better than the Therminol-VP1 oil.

Global, solar and thermal efficiencies were compared for the ET between the pressurized water, boiling water and Therminol-VP1 oil at different values of solar irradiance, where the efficiencies are increased as the irradiance increases.

Also it is concluded that the pressurized water has the highest efficiencies compared to the boiling water and Therminol-VP1 oil which has the lowest performance, for example at solar irradiance 0.70 kW/m^2 the solar efficiency when pressurized water is used equal to 64.99% while at the same point for boiling and oil it is equal to 59.89 and 56.86% respectively.

The highest values of global efficiency obtained were about 34.19%, 33.69% and 28.56% for boiling, pressurized and Therminol-VP1 oil respectively.

These results were compared with the results from literature for different types of parabolic trough, it is found that at the same value of $\Delta T/I_s$ for example 0.3 m^2K/W the solar efficiency of the LS-2 collector is equal to 50% which is smaller than the value that obtained for ET from the simulation (69.24%).

Also the thermal efficiency at $\Delta T/I_s$ = 0.2 m^2K/W was 52 %, and 51.5% for shielded class and unshielded class respectively, but for ET collector it is found about 51.84%.

Finally, Eurotrough collector was theoretically applied in different places in Jordan, where the performance of ET is carried out. Especially in Ma'an city the distribution of direct solar irradiance was calculated for different days around one year and the performance of ET presented as global, solar and thermal efficiencies was obtained.

6.2 Recommendations

After reviewing this study, several suggestions for further study have been formulated:

1. Minimizing the optical losses of the collectors' receiver.
2. Minimize the thermal losses of the collectors' absorber.
3. Decreasing the pipeline losses by finding the optimum plant layout and insulation requirements that give minimum losses from the pipelines.
4. Studying the properties of integration and optimization of the material of solar collector especially the coating and the heat transfer fluid.

5. All necessary techniques which could increase the temperature difference (ΔT) between the absorber outlet temperature (Tout) and absorber inlet temperature (Tin) are encouraged to be applied in the experimental study on the solar collector, for example, insulating the inlet storage tank and other relevant techniques.
6. The absorber of the new designed EuroTrough collector (ET) and also the existing parabolic collectors needs to be equipped with an evacuated glass envelope.
7. By designing, fabricating and testing the EuroTrough collector (ET), a comparison of test results between ET and the existing parabolic trough collectors (like LS-1, LS-2 , compound parabolic trough and LS-3) can be obtained.
8. Development of design and optimization techniques for the use of solar energy in cooling applications. This will involve the development of cooling cycles that partially or completely use solar energy. As such, the research will move from cogeneration to tri-generation.

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Appendix A: MATLAB Code and Sample of Results:

```
clear all
clc
%++++++++++++++++++++++++++++++++++++++++++
Tamb=30+273;
Pout=(200/8);
Is=0.3:.1:.8;
do=0.07;            l=12.27;
la=5.76;            pai=3.14;
%^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^
taw= 0.93;alpha= 0.95;row=  0.93;
Aa=  70.6752;Ar=2.69;CR=(Aa/Ar)*2;
epslon=1;segma=5.67*10^-8;
%^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^
C=0.53;n=0.25;g=9.8;B=207*10^-6;Kw=0.6;mu=1.983*10^-5;density=1.165;Dout=0.07;
mw= 0.038;l=12.27;L= 0.055;Twin=50+273;epslon=1;segma=5.67*10^-8;Dg=0.115;
Tamb=30;Tc=50+273;v=0.658*10^-6;Din=0.055;
Tw0=70+273;Two=Tw0:50:600;
%^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^
Ain=pi*Din*l;
Aout=pi*Dout*l;
Ag=pi*Dg*l;
V=mw/(density*Ar);
Re=density*V*Dout/mu;%Renolds number 'Re'
Pr=cpw*mu/Kw;%Prandtl number 'Pr'
Nu=0.023*Re^0.8*Pr^0.4;%Nusselt number 'Nu'
hin=Nu*Kw/L;%The convection coefficient between the water and glass cover 'hin'
%^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^
Tavgw=(Twin+Two)/2;
hr_c=segma*epslon.*(Tavgw.^4-Tamb.^4)./(Tavgw-Tamb);
%The radiation coefficient between glass cover and ambient 'hr_c'
%^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^
Gr=(g*B*(Tc-Tamb)*Dout^3)/v^2; %Grashof number 'Gr'
hout=C*(Gr*Pr)^n;%The natural convection coefficient between the receiver and glass cover 'hout'
%^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^
```




```
%*******************************************************
Gr=(g*B*(Tc-Tamb)*Dout^3)/v^2; %Grashof number 'Gr'
hout=C*(Gr*Pr)^n;%The natural convection coefficient between the receiver and glass cover 'hout'
%*******************************************************
Ul=1./((1/(hin)+Ain./(hr_c*Aout)+Ain/(hout*Ag))); %The overall heat loss coefficient 'Ul'
%********************************************************************
%for oil and pressrized
mw=(Pout/(1-(Tamb/Twout))/(cpw*(Twout-Twin))
%Twoutt=(taw*alpha.*row.*Aa.*Is-Ul.*Ar.*Tamb+mw.*cpw.*Twin-0.5.*Twin.*Ul*Ar)./(mw*cpw+0.5.*Ul*Ar)
%Twout=Twoutt+273
%mw=(Pout./(1-(Tamb./Twout))./(cpw*(Twout-Twin)))
%****************************************************
%for boiling
Twin=50+273;mw=0.038; Cpw=4.38
hin= 209.26
Is=[.2 .3 .4 .5 .6 .7 .8 .9 1];
hout=1.0e+003*[2.797 2.845  2.8750    2.9354    2.9621    3.0017    3.0337    3.075 3.0752];
%Twoutt=(((-mw*(hout-hin)+taw*alpha*row*Aa*Is+Ul*Ar*Tamb)/(Ul*Ar))-(397.177))/.225
deltaTw=Twout-Twin;
%********************************************************************
Aapr=la*l;
Tabsmean=.75*Twin+.25*Twout;
thermal_loss_coll_amb=Ul*pai*do*l.*(Tabsmean-Tamb)
eff_thermal=1-(Tamb./Twout)
```

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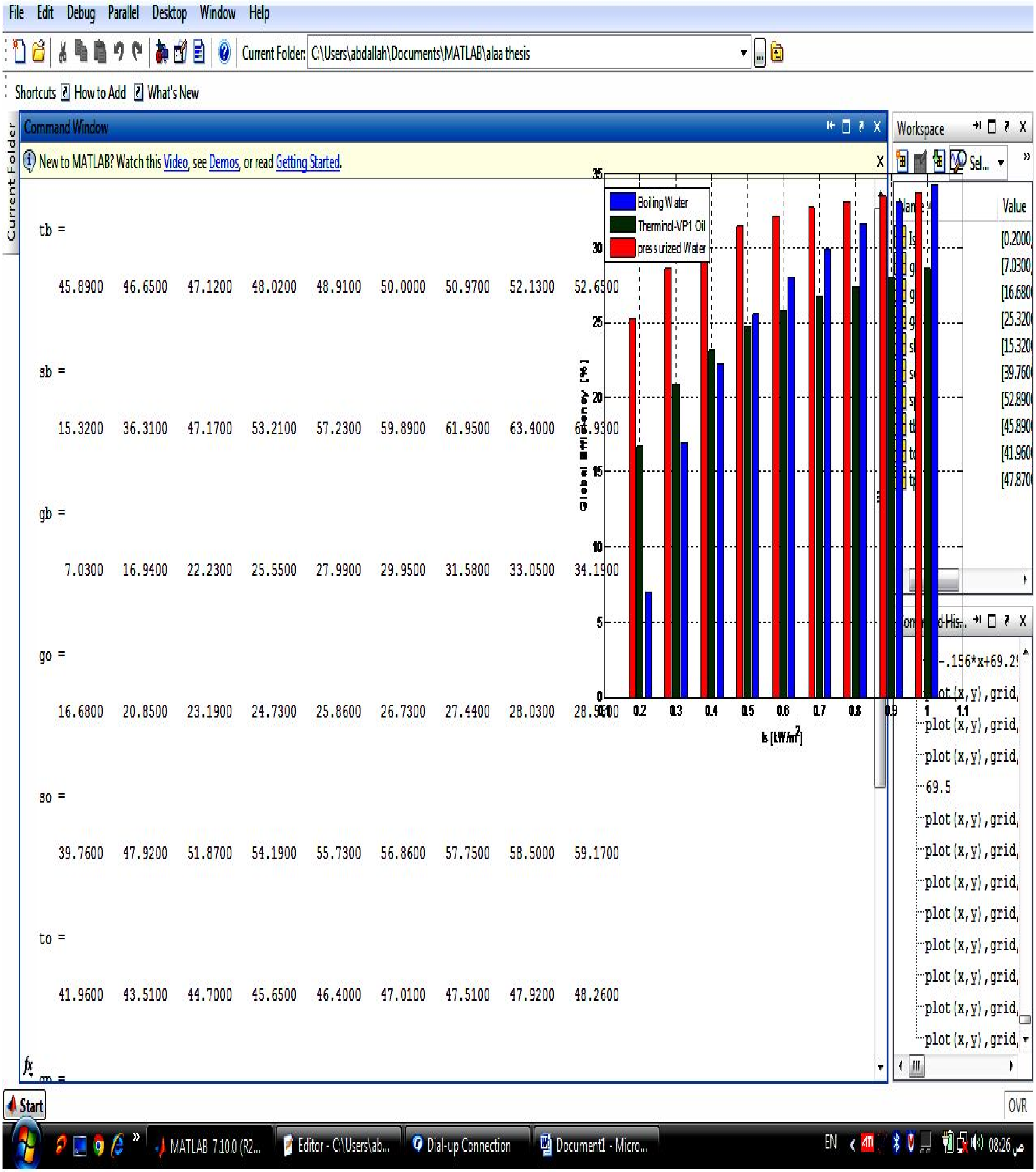
```
clear
clc
%Application of ET in Jordan
%Is calculation for different places in Jordan
%n=[17   47   75   105   135   162   198   228   258   288   318   344];
Gsc= 1367*.8;        omega=45;             ;phai=30.18;
n=228;
delta=23.45.*sind(360*(284+n)/365)
Is=Gsc.*[1+0.033.*cosd((360.*n)/365) ].*(sind(phai).*sind(delta)+cosd(phai).*cosd(delta).*cosd(omega))
%^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^
Tamb=30+273;     Twin=40+273;         Pout=(200/8);
do=0.07;            l=12.27;           la=5.76;
pai=3.14;
%^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^
taw= 0.93;alpha= 0.95;row=  0.93;
Aa=  70.6752;Ar=2.69;CR=(Aa/Ar)*2;
cpw= 1.968;%for oil
epslon=1;segma=5.67*10^-8;
Twoutt=(taw*alpha.*row.*Aa.*Is-Ul.*Ar.*Tamb+mw.*cpw.*Twin-0.5.*Twin.*Ul*Ar)./(mw.*cpw+0.5.*Ul*Ar)%for pres. and oil
deltaTw=Twout-Twin;
%^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^
Aapr=la*l
Tabsmean=.5*Twin+.5*Twout;
%thermal_loss_coll_amb=Ul*pai*do*l.*(Tabsmean-Tamb)
eff_thermal=1-(Tamb./Twout)
%mwl=(Pout./eff_thermal)./(cpw.*deltaTw)
eff_solar= (taw*alpha*row)- 0.01*Ul* (Tabsmean - Tamb)./(CR.*Is)
eff_global=eff_solar.*eff_thermal
```

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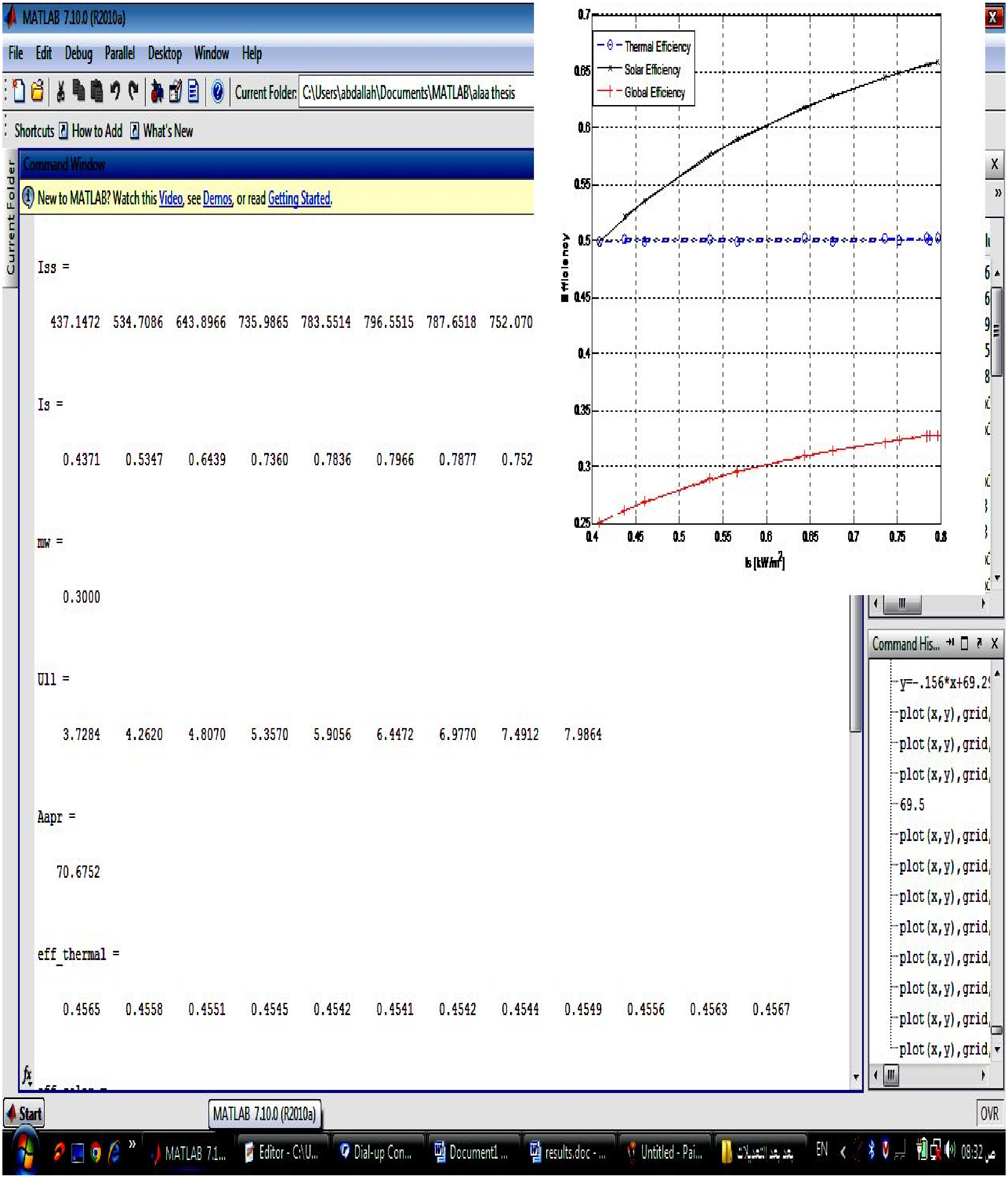
```
deltaTw=Twout-Twin;
%^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^^
Aapr=la*l;
Tabsmean=.75*Twin+.25*Twout;
thermal_loss_coll_amb=Ul*pai*do*l.*(Tabsmean-Tamb)
eff_thermal=1-(Tamb./Twout)
eff_solar= (taw*alpha*row)-Ul.* (Tabsmean - Tamb)./(CR.*Is)
eff_global=eff_solar.*eff_thermal
Poutt=Aapr.*Is.*eff_global
%plot(Is,eff_thermal,Is,eff_solar,Is,eff_global),grid,xlabel('Is %[kW/m^2]'),ylabel('Efficiency [W/m^2K]')
%plot(Twout,eff_thermal,Twout,eff_solar,Twout,eff_global),grid
%plot(Twout,thermal_loss_coll_amb),grid,xlabel('Two [K]'),ylabel('Thermal loss collector-ambiant [W/m^2K]')
%plot(Two,2*Ul),grid,xlabel('Boling Water Output Temerature (K)'),ylabel('Overall heat Cofficient (W/m^2)'),%axis([350 600 4 8])
%++++++++++++++++++++++++++++++++++++++++++++++++
%solving Twout, Ul
function s=solpropq(p)
[n,m]=size(p);
%x0=input('Enter the initial estimate -> ');
x0=[0;0;]
   iter = 0;                          % Iteration counter
c=zeros(n,1);
while abs(max(Dc)) >= 0.00001e-30       % Test for convergence
   iter = iter + 1;
   f=p;
jj=der;
   y=c-f;
   x1=x0(1);x2=x0(2);x3=x0(3);
  Dc=subs(y);
  J =subs(jj);
dx=inv(J)*Dc;
x0=x0+dx;
end
```

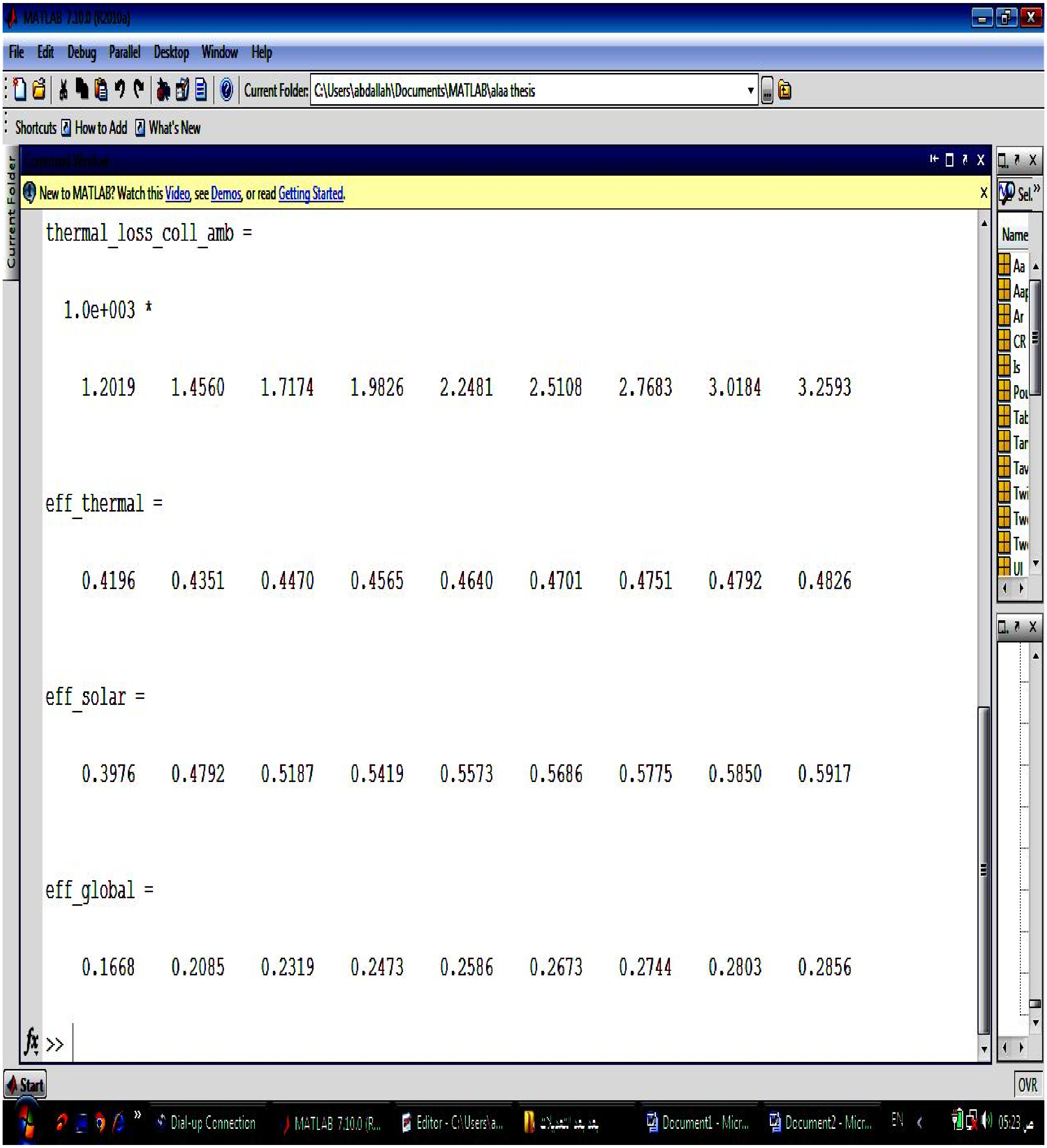
ET_simulation.m* | plotvaldiation.m | solpropq.m

script | Ln 82 | Col 1 | OVR



File Edit Debug Parallel Desktop Window Help
Current Folder: C:\Users\abdallah\Documents\MATLAB\alaa thesis
Shortcuts How to Add What's New
Command Window
New to MATLAB? Watch this Video, see Demos, or read Getting Started.
tb =
45.8900 46.6500 47.1200 48.0200 48.9100 50.0000 50.9700 52.1300 52.6500
sb =
15.3200 36.3100 47.1700 53.2100 57.2300 59.8900 61.9500 63.4000
gb =
7.0300 16.9400 22.2300 25.5500 27.9900 29.9500 31.5800 33.0500 34.1900
go =
16.6800 20.8500 23.1900 24.7300 25.8600 26.7300 27.4400 28.0300
so =
39.7600 47.9200 51.8700 54.1900 55.7300 56.8600 57.7500 58.5000 59.1700
to =
41.9600 43.5100 44.7000 45.6500 46.4000 47.0100 47.5100 47.9200 48.2600
Boiling Water
Therminol-VP1 Oil
pressurized Water
Global Efficiency [%]
Is [kW/m2]
Workspace
Value
[0.2000
[7.0300
[16.680
[25.320
[15.320
[39.760
[52.890
[45.890
[41.960
[47.870
-.156*x+69.2
plot(x,y),grid,
69.5
Start
MATLAB 7.10.0 (R2...
Editor - C:\Users\ab...
Dial-up Connection
Document1 - Micro...
EN
08:26


MATLAB 7.10.0 (R2010a)
File Edit Debug Parallel Desktop Window Help
Current Folder: C:\Users\abdallah\Documents\MATLAB\alaa thesis
Shortcuts How to Add What's New
Command Window
New to MATLAB? Watch this Video, see Demos, or read Getting Started.
Iss =
437.1472 534.7086 643.8966 735.9865 783.5514 796.5515 787.6518 752.070
Is =
0.4371 0.5347 0.6439 0.7360 0.7836 0.7966 0.7877 0.752
mw =
0.3000
Ull =
3.7284 4.2620 4.8070 5.3570 5.9056 6.4472 6.9770 7.4912 7.9864
Aapr =
70.6752
eff_thermal =
0.4565 0.4558 0.4551 0.4545 0.4542 0.4541 0.4542 0.4544 0.4549 0.4556 0.4563 0.4567
Thermal Efficiency
Solar Efficiency
Global Efficiency
Efficiency
Is [kW/m2]
Command His...
y=-.156*x+69.2
plot(x,y),grid
69.5
MATLAB 7.10.0 (R2010a)
Start
OVR
08:32


MATLAB 7.10.0 (R2010a)
File Edit Debug Parallel Desktop Window Help
Current Folder: C:\Users\abdallah\Documents\MATLAB\alaa thesis
Shortcuts How to Add What's New
New to MATLAB? Watch this Video, see Demos, or read Getting Started.
thermal_loss_coll_amb =
1.0e+003 *
1.2019 1.4560 1.7174 1.9826 2.2481 2.5108 2.7683 3.0184 3.2593
eff_thermal =
0.4196 0.4351 0.4470 0.4565 0.4640 0.4701 0.4751 0.4792 0.4826
eff_solar =
0.3976 0.4792 0.5187 0.5419 0.5573 0.5686 0.5775 0.5850 0.5917
eff_global =
0.1668 0.2085 0.2319 0.2473 0.2586 0.2673 0.2744 0.2803 0.2856
fx >>
Start
OVR
Dial-up Connection
MATLAB 7.10.0 (R...
Editor - C:\Users\a...
Document1 - Micr...
Document2 - Micr...
05:23

تشبيه ودراسة أداء لاقط شمسي مجمع متطاول من نوع القطع المكافئ

اعداد
آلاء محمد القطيمات
المشرف
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ملخص

أدت زيادة تكاليف الوقود الأحفوري في العالم لاستخدام الطاقة الحرة والمتاحة بشكل طبيعي من الشمس لانتاج الطاقة الكهربائية .الاقط الشمسي المجمع المتطاول من نوع القطع المكافى في الوقت الحاضر هو النظام الأكثر استخداما الطاقة الشمسية لإنتاج الكهرباء واستخدامها في مولدات البخار في العمليات الصناعية . ويعتبر من انظمة التكنولوجيا الطاقة الشمسية الاكثر ثباتا والاقل تكلفه المتاحه حاليا على نطاق واسع .وتتألف اساسا من المجال مجمع الذي يحول الإشعاع الشمسي إلى طاقة حرارية والحرارية التقليدية الى طاقة كهربائية لدورة رانكين .في هذه المصانع ، يمكن تنفيذ نظام تخزين من أجل زيادة الإنتاج .

هذا العمل يهدف إلى إيجاد نموذج محاكاة لمحطة الطاقة الحرارية الشمسية ويقترح استخدام اللاقط الشمسي المجمع الاوروبي من نوع القطع المكافئ باختياره لتجميع الطاقة الشمسية. يستخدم برنامج ماتلاب لتقييم الأداء وتحليل السوائل المختلفة في المحاكاة هي : المياه المضغوطه والمغلية ، والزيت (ثيرمينول) وعملت مقارنه لنتائج الاداء بينهم. وقد وجد أن استخدام المياه (الضغط أو المغليه) في أنبوب المتلقي هو أفضل من الزيت (ثيرمينول) .والمياه المضغوطه لديها أعلى قيمة من الكفاءة مقارنة مع الماء المغلي والزيت (ثيرمينول) الذي لديه أدنى الأداء .كما تم تطبيق أداء اللاقط الشمسي الاوروبي على أماكن مختلفة في الأردن ، وتوزيع الإشعاع الشمسي المباشر في أيام مختلفة على مدار سنة واحدة تم حسابه لمدينة معان.